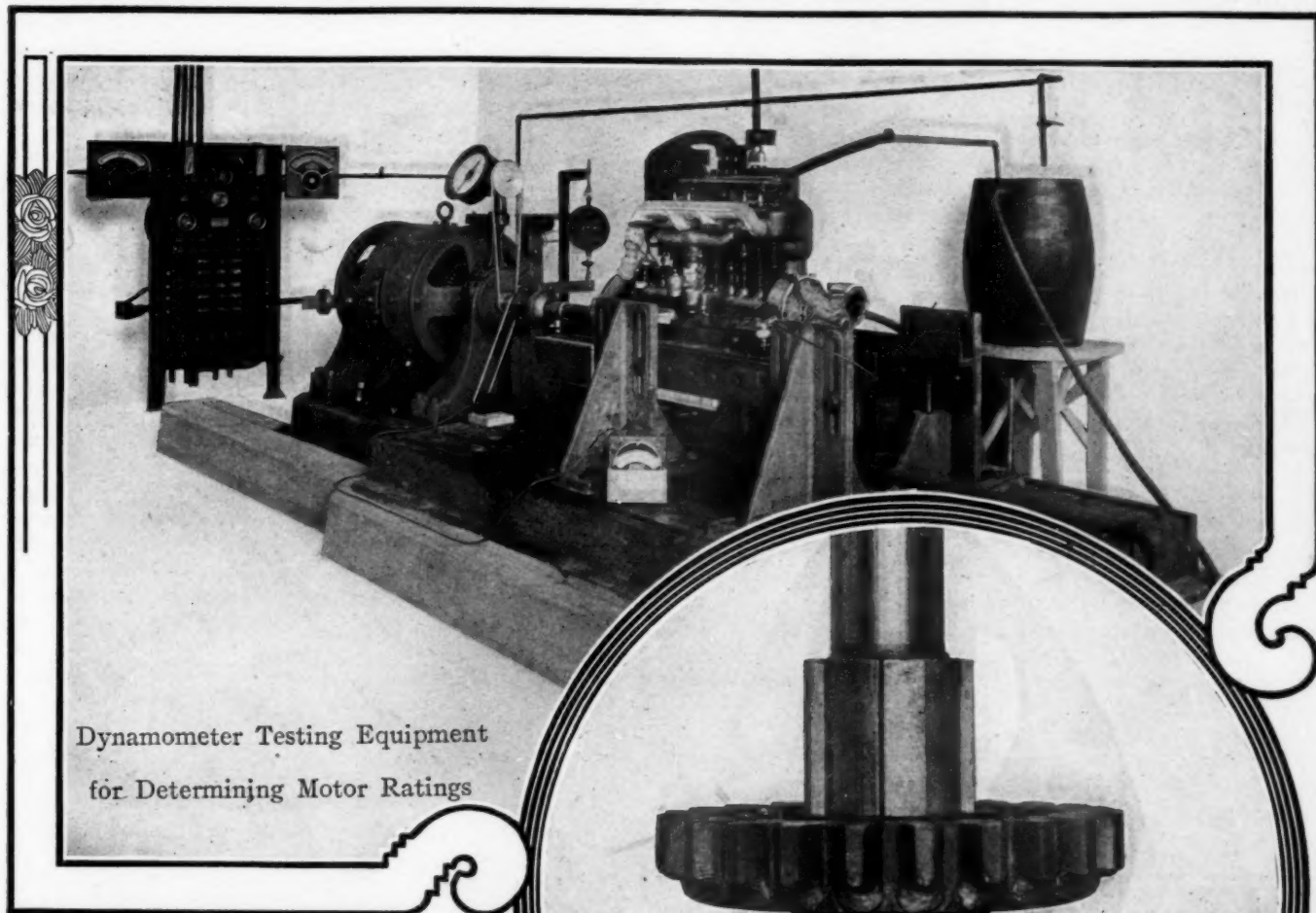


# THE AUTOMOBILE

## Calling Attention to Major Problems Differentiating Between the Types of Undertakings



Dynamometer Testing Equipment  
for Determining Motor Ratings

*On the same basis that the books in a library are classified and indexed, it is suggested that the good, the mediocre, and the bad should be indexed also, it being the idea to support the wise proceedings by suitable additions thereto, and to annul the influence of the mediocre by wiping transactions of this character off the slate, excluding the matters of a low order of merit altogether.*

**E**NGINEERING WEEK in the automobile industry is the time when the brains of the art assemble for purposes of discussion, reviewing the doings of the past, and marking out the working plan of the activities of the near future.

Closer Limits  
of Tolerance  
Required in  
Practice

Example of a  
Poor Fit of  
a Repair  
Part



To count the cost and obtain the means for the end is a part of the undertaking.

During the last year, the Society of Automobile Engineers has directed its effort to the classification of the raw materials that are used in the making of automobiles, paying proper attention to the quality thereof, with particular reference to the commercial sizes of the things that have to be employed, limiting the number in each division to the minimum demand, thus saving in cost, and in the time it takes to get material and to whip it into shape.

But these are A, B, C activities from the point of view of the future, nor do they represent in a direct sense the things that should be done in order that the purchaser's investment may be conserved. It is a matter of no moment at all how good an automobile may be if the purchaser is confronted by an unknown future, for in all fairness, the initial quality of a car is reflected in the initial price, and the cost of maintenance thereof will be out of all proportion if the automobile is good and the price is high to accord therewith, provided repair parts have to be whittled out by hand in a garage repair-shop in order to keep the automobile on the road after it shows the mark of service and of time.

Admitting, then, that the engineers should tell the makers of automobiles how to assure to themselves a liberal and lasting dividend in payment for their enterprise, this is not to say that these engineers should exclude from their thoughts the further advantages that will accrue to purchasers if an organized effort is made to prolong the life in service of the automobiles produced, minimizing cost, and saving upkeep troubles. It would appear, all things considered, that the engineers of the industry have larger duties before them than the measure of recognized responsibilities, among which mention will be made here of (a) a plan of "service," founded upon the idea of promptly supplying repair parts at moderate cost to more or less worn-out automobiles, at any time within ten years after the automobiles are sold; (b) the establishing of a general laboratory for the common good of all the makers of automobiles, and the edification of the purchasers of cars, and (c) the elimination of the annual model practice, for the good of the industry.

#### How Classification of the Phases of Automobile Making Would Redound to the Advantage of All Concerned.

Taking up with the question of "service," remembering that an automobile must wear out in the regular course of events, and that its measure as an investment depends upon the cost of upkeep, it will be seen at a glance that a relatively inferior type of automobile would take rank as a superior investment were it possible to get repair parts promptly at a low cost at any time within ten years after the automobile is sold, were we to compare this idea with that which obtains in connection with the purchase of a superior make of car, if of the latter it might be said, repair parts are not to be had at all, through the ramifications of a classified process.

From the point of view of a laboratory, it has been said on repeated occasions by automobile engineers who assembled to discuss such matters, that no individual maker of automobiles can afford to establish a fully equipped laboratory and maintain the same in constant service, for no better purpose than to make experiments, and yet these experiments, so called, must take rank in the long run as mere efforts to get at the facts attending much-needed and valuable information.

Of the several attempts to establish laboratories on an independent basis, little can be said of them excepting that they were in the right direction, but the men who worked upon this plan realized from the start that the unification of the automobile and its component parts will only come in time, and as the product of unbiased effort on the part of men who may be permitted to devote themselves to the task unhampered by dividend considerations.

Of the many things that must be done on a laboratory basis, there is nothing that promises more than "motion studies," coupled with the fixing of limits of tolerance, it being the case that the amount of interchangeable work that can be produced

in a given plant would far exceed the most sanguine expectations of "accelerators," as they obtain at the present time. In the title illustrations, in addition to presenting a testing equipment for determining the capabilities of motors, carbureters, magnetos, conditions of lubrication, effects of cooling, etc., a second illustration is presented showing a sliding gear on a shaft placed in the vertical position to indicate that the gear sticks, due to lack of clearance, but this example of a malfit is but one of many. The reason for making this point at this time is to bring out a more exacting situation that has received but small consideration at the hands of designers. It is common practice in the plants to establish clearance without taking into account the ills of torsion, and if we may turn the process in vogue as a "static" method of fixing limits of tolerance, it will at once appear that this method is entirely wrong, and that the kinetic conditions must be taken into account before the effort can be looked upon in the light of a sanctioned process.

But in the fixing of the limits of tolerance along kinetic lines, it must be remembered that the influence of torsional and bending moments is magnified as the members are increased in length for a given diameter, and if we inspect the formulæ that are looked upon as rational for the conditions named, it will then be seen that the limits of tolerance must be varied, not in proportion to diameter, which is present practice, but in the light of length, considering diameter, not forgetting the influence of torque. In the example of the sliding gear on its splined shaft as shown in the title illustration, the gear was originally provided with a sufficient clearance under static condition to serve every apparent purpose, but in service the shaft twisted and the amount of the twist was sufficient to induce a condition of permanent set after which the gear stuck and ceased to be of value for its intended purpose. This is a very common type of trouble that has recurred in automobiles hundreds of times within the last ten years and many engineers linger on the brink of uncertainty, wondering whether or not they will make these shafts of such a considerable diameter that they will not twist at all, or make the clearance so great that the gears will slide with freedom after initial set is induced.

Taking this simple situation as the foundation for the thought that a laboratory might be engaged advantageously over a period of years in the solving of many problems, it is only necessary to enlarge for a little more, bringing out the idea that there is a fallacy concealed in the thought that poorly fashioned members will work satisfactorily if the quality of the steel is improved, as when alloy steel is employed, in view of the fact that the modulus of elasticity is scarcely any better for the finest grade of alloy steel than it is in a normal grade of Bessemer bar. From the point of view of torsion, if the modulus of elasticity is constant in steel of all grades, the "rational" formula may still be used to direct attention in safe channels, and the limits of tolerance must then be fixed to satisfy these formula.

Motion study investigations cannot be conducted satisfactorily without working on the limits of tolerance that should be set for the work at the same time, in view of the fact that in fixing limits of tolerance, it is for the purpose of aiding the workmen who would take a very long time doing an operation to dead accuracy, and who would take a relatively small amount of time performing the same operation within reasonable limits of accuracy. It has been found in practice that parts will be quite interchangeable if a system of tolerance is employed, whereas the idea of working to a "neat" fit is impossible.

Finally when automobiles are made and they are placed in the hands of the testers whose duty it is to tune them up, it will be worth more than a little to make them keep accurate records of what they do; otherwise, the faults of construction, if such there are, will remain in the automobiles for all time. Again, in contest work, to get rid of the irresponsible type of publicity man, and to place the work under the charge of men of stamina, is to get the best result; and, strange to relate, it is the best way to get favorable mention in the columns of the press, this being desirable.

# In the Rating of Automobile Motors

## Intricacy of the Problem Appreciated Abroad

*Rating motors from the point of view of the engineer in the shop is beyond the reasoning power of the casual observer whose closest familiarity with the automobile problem is confined to the use of a car in service, it being a condition of this use that the power of the motor is unimportant from the point of view of service as long as the car runs and satisfaction is delivered to the operator. It is related in the story how automobile enthusiasts busy themselves to no purpose nearly all of the time, but when they are confronted with the necessity of buying an automobile they fail to follow the bent of their own musings, and if they get the kind of a car that is suited to their need it spells nothing by way of foresight on their part.*

WHEN a committee of the Mechanical Branch of the A. L. A. M. fixed upon a formula for use in denoting the horsepower of automobile types of motors, ending by putting its stamp of approval upon the single algebraic statement:

$$H. P. = \frac{d^2 \times n}{2.5}$$

The committee was mindful of the capacity of the layman for knowledge of this type, although, as after events would seem to prove, the committee shot too high, and probably 50 per cent. of all of the laymen who deigned to take notice of matters of this sort are wondering what it is all about.

In the face of this difficulty it would seem as if the committee relied too much upon its audience, or, better yet, the presence of an audience, and the idea of erring in the direction of simplicity at the expense of accuracy for no better purpose than to compel men who do not wish to understand to go into a matter that does not concern them overmuch, must be measured on the scale of inefficiency. Users of automobiles are justified in taking the ground that they are not interested in the abstract ability of automobile types of motors, provided, of course, that the particular motor used in a given case is in harmony with the requirement.

The effort required to propel an automobile on a conventional road depends upon the weight and speed of the car, and the motor that will do this work satisfactorily to the owner of a car may be styled by any name whatsoever without disturbing the equanimity of the man whom it is intended to serve. If it was the idea of the committee to furnish a basis for the naming of motors there was merit in the proceeding, in which event a concerted effort on the part of some body of competent jurisdiction that will make for a uniform practice will be worth while. But if it is the contention on the part of engineers that motors should be given a rating in horsepower, and that purchasers should pay attention to the horsepower ratings of motors that are used in the cars of their choice, it is then proper to observe that these ratings should be in fact rather than the plaything of the publicity man who will not hesitate to make capital out of the fact that real engineers are responsible for the rating and that it must be so.

It is not the purpose here to say that a motor of the type as used in automobile practice may not deliver power according to the formula as given, but it is true that there is no

reason why a motor should live up to this formula, and it will be a coincidence if it does. It is no argument to say that the real situation cannot be disclosed to the audience because of the complexity of the problem and the obtuseness of the audience. If a poem is original in the Greek tongue and the beauties of its meter are to be extended and laid bare for the edification of the largest audience, there would be no sense in butchering the poem and rendering a version of the same that would be deprived of the beauties of the original and which would fall upon the senses of the victim like a pall. The actual horsepower rating of automobile motors has all the qualifications of a beautiful Greek poem, and the understanding beats against the wall that lack of knowledge has built up around this important matter, leaving it for some body of men to struggle with the intricacies of the undertaking, not for the purpose of allaying the fears of the obtuse, but with the understanding that the facts will come out and be used in the course of time for the benefit of the automobile industry.

In the building up of the literature of the automobile business, if we go on presuming that things are so and so in the absence of information that would establish a line of practical facts, the whole structure will be on a parallel with the fabric from a literary point of view of the steel industry, in which it is not difficult to hear a purchasing agent discuss the poetry of steel, enlarging upon the qualities that are said to reside in "jewelry" product, and as he rests, or better yet, pauses for breath, he takes the spare moment to fill out an order for 20,000 tons of cold-rolled steel, thus traversing the hills and the vales of the long road that leads from the sublime to the ridiculous.

### There Is No Sense in Trying to Fasten a False Standard Upon the Users of Automobiles When Their Need, if Satisfied, Terminates the Undertaking

Having arrived at the conclusion that a problem of the intricacies of the one in question is far too academic to be inflicted upon a mere purchaser of an automobile whose want will be satisfied if the car will do the work for which he buys it, the ground for delving in simplicity at the expense of accuracy is cut from under, and the engineers in America are given a free field with nothing before them but to get at the facts and use them for what they are worth.

If it can be shown that all motors are exactly alike from the point of view of power and weight efficiency, and if it may be said that these properties are maximum as they reside in motors as actually made, it is even a question as to whether or not the engineers of the industry will have to consider it worth their while to struggle with the problem since they would have nothing to gain thereby. But the situation is quite different; the motors are not all alike. There is a distinct dissimilarity between them as respects their performance, and in a hundred ways, perhaps, the divergencies fall below the maximum practical possibility, thus showing that lack of knowledge is having its influence, and that a standard to go by is essential to the unification of the motor-building art, just as a wheelman needs a "pacer" in order that he will be able to tell whether or not he is doing his best in a given endeavor.

It has been said that trade jealousies interfere with co-operative projects, but wisdom coupled with perspective decries this idea, and certainly the perspective of the automobile business shows that men do emulate good example, and this point

is fittingly brought out by the uniformity of practice that is to be seen at every hand, making the automobiles of to-day so much alike that it takes an expert to describe their differences, and so different are the cars of the present time from those of yester-year that a blind man could distinguish between them by the mere touch.

But if good is to come in the process of taking advantage of the merit that resides in the examples of a like art, as they are to be seen in the surroundings, let us not overlook the fact that the greatest measure of good will follow in the shadow of prompt action. If we must copy each other let us do it with systematic promptness. And if we admit that copying is a profitable thing to do let us help each other in order that the number of mistakes will be reduced to a minimum and the copies of good examples will be obedient children, bearing a fair resemblance to the sources of their inspiration.

It is a strong temptation to put forth the claim of originality if a plan works well, but after all, the efforts of men are not different from the translations that other men have made, and it may not be out of place to observe that when a translator is commended for the good work that he does in rendering a classic intelligible in another tongue he is not looked upon in the odious sense that is inferred when reference is made to a man as a "mere copyist." There is nothing of merit in the copying of an idea that is devoid of merit. On the other hand, it is a high intellectual attainment to be able to discriminate between the good and the bad, to separate the one from the other, and, furthermore, to take advantage of the fine qualities of the praiseworthy in whatever field of endeavor it may be found.

Commercialism does battle with intelligence to whatever extent it depreciates effort of the type that has to do with the emulation of a good example. The man who makes the first copy of an excellent idea, or who has the good sense to combine a plurality of good ideas, rendering them harmonious, is too prone to look upon himself as the originator of a great good, and in the egotism of his mind his nimbleness is rendered potent in the direction which does not indicate his high appreciation of a similar effort on the part of a brother workman in the vineyard.

In patent law there is a tradition to the effect that an old thing applied to a new purpose constitutes invention. Disregarding the further ramifications of patent law, this is another way for saying that a good idea can be distinguished from a mediocre proceeding, but we are prone to attach a surfeit of significance to this patent proposition, and of the inventor, so called, it may be said he is inclined to limit the dividend that would come to him in the natural course of events as the product of his fertility through his desire to stand on a pedestal unattended by any so great as himself, according to the state of his mind, and in this condition of isolation the world takes

small heed of the character of the greatness involved, and the invention, in addition to being good, has the further property of self-strangulation.

As a fitting illustration of the futility of locking up good ideas, the present status of the steam automobile may be cited, in which, despite the excellence of the plan, its commercial growth has been stunted, and in the

race for supremacy it is conspicuous for its fine performance in the few examples that have to be sought for to be found. Who can tell what would have been the history of the steam automobile in the absence of the shackles that debar engineers from taking advantage of the well-known facts in relation to its exploitation? The gasoline type of automobile has resisted the onward march of progress at every turn, due to the inherent difficulties involved in the building of gasoline motors, augmented by the fact that there were no men skilled in the building of variable-speed internal combustion motors, even ten years ago, whereas in the steam art there were thousands of capable engineers and among the supporting clientele of the automobile industry it would have been difficult to find a purchaser who would confess entire ignorance of the ramifications of steam. The whole history of the steam automobile is a staunch argument in favor of co-operation of the builders of the types of automobiles that are now current, and the quicker we reach the conclusion that it is better to copy the meritorious ideas from whatsoever source the greater will be the total of the gain, under which conditions the share that will accrue to the respective members of the fraternity will be perceptibly more; hence the wisdom of the plan.

### The Greatest Effort Will Be Required in the Process of Whipping the Motor Into Line

That there is no end of illustration of the wisdom of co-operating in a big undertaking is shown by the diversity of the mediocre result that is produced in the absence of team work. Narrowing the discussion down to the problems involved in the motor confronts one with the enormity of the undertaking, and the first estimate leads to the conclusion that no one maker of automobiles can afford to run a laboratory long enough to thrash out the details of a working plan. The old

idea of the Mechanical Branch of the A. L. A. M. had its measure of merit, due to the fact that many investigations could be carried on under the direction of men of accented skill, with the cost to be borne by a considerable number of companies, limiting the increment to the respective concerns below the point where it would have any

marked effect upon the "overhead" of the confederating companies in their individual aspect.

Whether or not the time will soon come when the Society of Automobile Engineers will have money enough in its treasury to conduct an independent series of investigations along the necessary lines is a problem that is too intricate to be solved offhand. Whether or not it would be desirable for the society to take money from the makers of automobiles and use it in this enterprise is a matter that has an ethical side, and it, too, partakes of complications that are not to be disposed of at a moment's notice.

In the meantime it may not be out of place to recite a few of the things that rest in the shadow of uncertainty, as, for illustration, we know that the thermal efficiency of a motor decreases with the amount of heat that traverses the cylinder walls and is taken up by the cooling solution and finally dissipated in the air that is pumped into the radiator through its interstices and at the loss of some power in the doing thereof. It is a clear case of paying the cost of doing the thing that

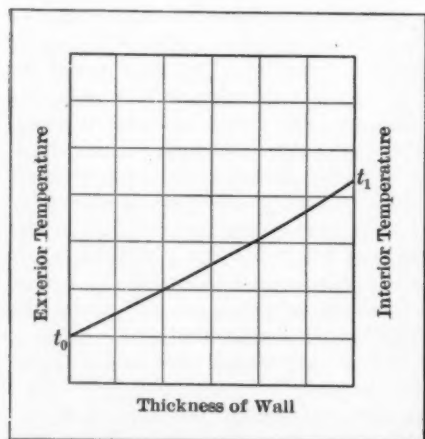


Fig. 1—Showing the interior as compared with the exterior temperature of a cylinder wall during the exhaust stroke

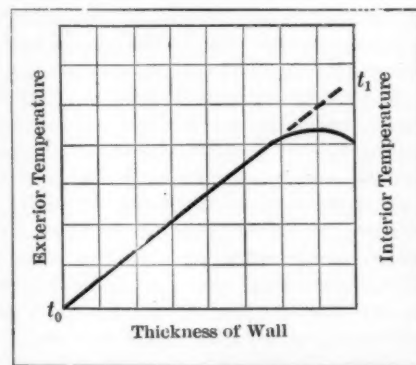


Fig. 2—Showing the interior as compared with the exterior temperature of a cylinder wall during suction and compression strokes

detracts from the value of the main effort, and we know from experience already gained that the amount of heat which will go through the cylinder walls depends upon the area of the flame-swept surface as a primary consideration. Limiting the area of the flame-swept surface seems to be a step in the right direction, and yet there is a diversity in practice that cannot be accounted for excepting on two counts, one of which has to do with the knowledge of the designer in individual cases and the other suggests that what we know is not so. If, for the sake of argument, it is assumed that the losses through the water jacket will be a minimum if the area of the flame-swept surface is reduced to a minimum also, we are still confronted with a problem that has never received enough consideration to dignify it with the appellation "investigated." In a word, having limited the amount of heat that can get through

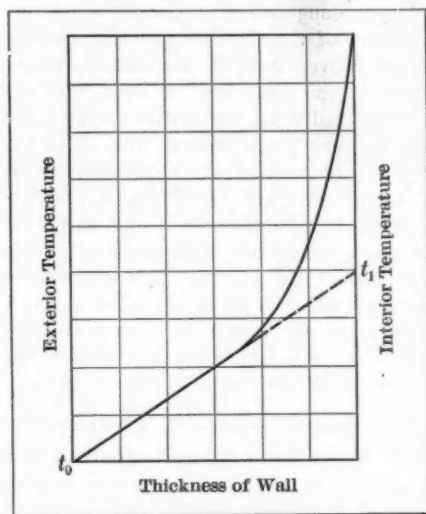


Fig. 3—Showing the interior as compared with the exterior temperature of a cylinder wall at the moment of explosion

the wall of the inner chamber, it remains to struggle with the passage of that heat through the metal of the cylinder and its transfer to the water of the jacket, and we know from experience that the temperature of the metal of which the cylinder walls are composed increases because the rate of transfer of the heat through the metal is slower than the rate at which the same heat passes into the body thereof and the accumulation of this heat in the metal composing the wall complicates the performance of the motor, resulting in the lowering of the thermal efficiency, not by direct heat loss, but due to the fact that the heat thus stored in the metal during the power stroke is given back to the relatively cooled incoming gas on the succeeding suction stroke, and the gas thus heated swells so that the quantity thereof that is taken in during inspiration is less than it ought to be. Admitting that there is no evidence such as will permit of fixing upon the values of the heat relations that obtain during the strokes of the cycle, the fact remains that it is possible to indicate the trend as it obtains in practice, and what is wanted in the long run is a series of investigations that will lead up to a better measure of more exact knowledge. As a general proposition the heat transfer through the cylinder walls causes a rise or fall in temperature, under the conditions as follows: Referring to Fig. 1 during the exhaust stroke, the temperature is  $t_1$  at the beginning of the stroke, and it falls to  $t_0$  with considerable rapidity. In Fig. 2, during the suction stroke the temperature residing within the cylinder is  $t_1$  at the beginning of the inspiration, but the exterior wall of the cylinder at this instant is given the value of  $t_0$ . At the moment of explosion the diagram, Fig. 3, is characteristic, and the temperature that resides within the cylinder raises from the value  $t_1$  for the dotted line to the considerably higher value as indicated by the reach of the solid line, and the temperature of the exterior of the wall during this time is given the value of  $t_0$ . During the power stroke the interior temperature gives a complex shape to the curve, and the value  $t_1$  is assigned to the point of the crossing of the solid with the dotted line, and in this case, as in the preceding examples, the simultaneous exterior temperature as measured on the wall is assigned the value  $t_0$ .

Granting that it is desirable to prevent the transfer of heat

from the inner surface of the combustion chamber to the metal of which the wall is composed, but remembering that this is an impossible condition to impose, it remains to be said that the next requirement is in the direction of a type of metal that will permit the heat to traverse the thickness of the wall at the same speed that it penetrates the interior surface, so

that the temperature  $t_1$  will remain at the same value as the temperature  $t_0$ , thus preventing the accumulation of heat in the metal of which the walls are composed, so that losses, to whatever extent they are unavoidable, will be primary in nature, and the compounding thereof will be thwarted. The diagrams Figs. 1, 2, 3 and 4, show at a glance that the ills of a sluggish transfer of the heat to the metal of the walls are minimized if the walls are thin, and this brings us to an accidental advantage that came to the designers of motors for automobile work, it being the case that the walls of the cylinders of automobile motors are made thin to eliminate weight, and we doubt if the designers of these motors realized that they were making high speed possible by the very process that called for the thin walls. If it may be remembered, the designers of stationary motors, prior to the coming of the automobile, were wont to provide a large factor of safety in the thickness of the walls, and by so doing, making the walls more or less three-quarters of an inch thick, they placed a limit upon the attainable speed of their product, due to the fact that the heat which accumulated in the section of the metal put a brake upon the performance, and it was a very excellent design of stationary motor that would run successfully at 350 revolutions per minute, largely on this account.

The use of cast iron in cylinder work puts a stop on the idea of making the wall thickness less than 6 millimeters, but there is room for investigation along lines for the purpose of determining whether or not the addition of copper, for instance, to the cast iron will help in the transfer of heat through the wall of a given thickness; in other words, we have yet to learn the extent to which a change in the chemical composition of the iron will have a favorable influence upon the transfer of heat through the wall, and this is beside any question of the use of some metal other than cast iron, with the idea that the walls can be made thinner, for no better purpose, perhaps, than to hasten the transfer of all of the heat that penetrates into the metal, with the intention of preventing a rise in temperature such as will react unfavorably, and among the difficulties involved reduce the actual weight of the incoming mixture, which is another way for saying that the weight efficiency of the motor will be reduced, not forgetting that the thermal efficiency is a variable, and it is likely to be on a decreasing basis if the weight efficiency falls down.

#### Problems Involving the Mixture, Compression and Burning of the Fuel

Owing to the practice of having carbureters, magnetos, and other functional accessories made without considering the particular motors on which they are to go, a certain lack of co-ordination creeps in, and it leads to the discussion of the carbureter problem on the one hand, the ignition ramifications as a secondary matter, and the motor performance without analyzing in detail the specific relations of the attending members on

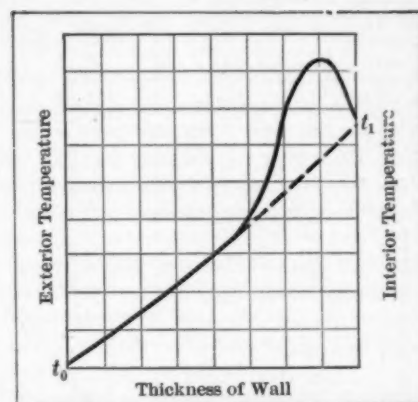


Fig. 4—Showing the interior as compared with the exterior temperature of a cylinder wall during the power stroke

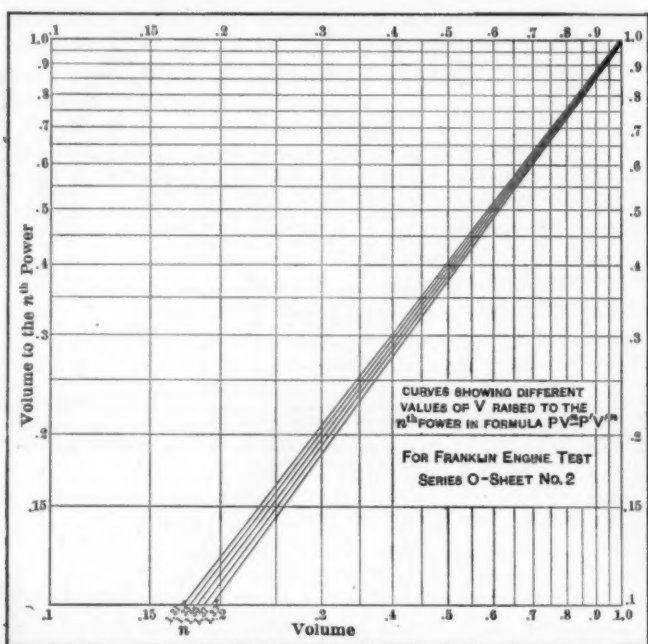


Fig. 5—Chart showing values of the exponent "N" under varying conditions of volume for different values of  $V$  raised to the  $n$ th power in the conventional formulae

an intimate basis. It is all very well to talk about the fine qualities that reside in a given make of magneto, but they will fail to support the issue if the magneto is put on a motor under conditions that will not permit of its effective use. To get the force of this presentation, it is only necessary to point out that a magneto, however good it may be, that is designed for a 25-degree spark advance may serve efficaciously on a motor that will thrive when the spark is advanced 35 degrees, but this magneto will fail to serve an exacting purpose if it is put on a motor that requires a 60-degree spark advance. If it is the idea of the audience that there can be no such wide variation of the spark advance requirement, all the audience will have to do for its proper edification is to examine one or two good makes of two cycle motors, trying them out first with the magneto ignition limiting the spark advance to say 35 degrees, and then make a comparative trial, using the same motor equipped with a magneto that will afford a spark advance of 60 degrees.

Granting that the magneto problem is given its measure of attention in a given case, and that the magneto employed is capable of delivering a spark of considerable energy, with a break-down voltage of perhaps 30,000 volts, the designer of acumen is then at the threshold of his trouble; he will be unable to use this wonderful magneto, getting all there is in it out of it, due to the fact that the spark plugs will break down at perhaps 2,000 volts, although it will be quite possible to get a considerable number of spark plugs that will stand up to 4,000 volts, and there will be a few examples of these devices that will hold out up to 8,000 volts, and after a long search perhaps one example may be found that will not break down inside of 16,000 volts, but it would be almost a vain search to look for a spark plug that will resist the wiles of 20,000 volts.

But even if the magneto and the spark plugs as they are employed in a given case are in some harmony with each other, there still remains the wiring system to be unified, and remembering that a magneto delivers a high potential alternating wave of electromotive force, and that inductive disturbances are absolutely assured, we are confronted by a hundred possibilities that we know nothing about. It is admitted that the high tension cable that is given a good coat of efficient insulation will stand up to something like 28,000 volts, where the insulation is unimpaired, but there are a considerable number of joints in an installation of this character, and of them it may be said, the insulation is far below that of the unimpaired section of the

rubber insulation on the new cable. Then comes the ills that may be traced to inductive disturbances, and if a metal tube is used, and self-induction is induced, the effectiveness of the wave of electromotive force that is set up in the windings of the armature of the magneto may be much reduced. None of these questions infringe in any way upon the troubles that may come in the course of service, due to deterioration of the magnetic properties of the permanent magnets of the magneto, nor has any mention been made of the hysteresis losses that may be due to the use of poor metal in the rotor of the magneto, or to the fact that time and service is a sufficient ground for change in the magnetic properties of the material, but it would be interesting to know whether or not the magneto used in a given case takes advantage of the special types of "transformer" iron that are designed to resist deterioration so that the hysteresis value of the metal under such conditions remains at a nearly constant level for a long period of time.

In the further consideration of the ignition problem, remembering that a magneto must deliver from its secondary winding a high electromotive force, it suggests the thought that the diameter of the magnet wire used must be relatively small, and that the  $I^2R$  losses must be very great indeed, and since the formula shows that these losses increase as the square of the current in amperes, are we not justified in arriving at the conclusion that a magneto is likely to burn out if the speed increases beyond a certain point, unless the designer thereof has contrived in some way to thwart the increase in electromotive force that is the normal expectation from an increasing speed, thus automatically preventing the electromotive force from raising beyond a certain point, and saving the burning out of the magneto by limiting the current in the natural order of things?

#### Should the Automobile Designer Make His Own Deductions or Rely Upon Magneto Engineer to Do His Thinking for Him

Take for illustration the body politic of the automobile engineers in America; what one of them would be willing to stand up at a meeting and say: "I think so little of my ability to unravel a tangle that I am willing to let some other more fortunate individual perform this function for me!" It is highly improbable that there are any candidates for a position of this sort, and yet the effect is the same. But in the long run, in the unifying of the conditions that make for quality in a motor, every important detail from the intricacies of magneto designing to the fallacies in spark plug work, must be brought to the light of day and examined with the utmost care by every man who styles himself a designer, and who is given the responsibility of such an undertaking. It was only the other day in a race that an automobile went out of commission because the magneto burned out and remembering that this magneto was probably one of a considerable number that are being used on actual automobiles, the thought comes that the makers of automobiles should instruct their designers to look into the matter and get, if they can, the kind of magneto that will have a struck balance of the heat losses in the windings against the heat emanations from the surfaces thereof.

In the designing of a magneto, or for that matter, any type of electric generator, it is a recognized fact among those who cope with such problems that the surface for radiation must be some safe proportion in view of the losses that are sustained under the most severe working conditions. In any winding, the loss in watts is equal to  $I^2R$ , remembering that  $I$  is the current in amperes and  $R$  is the resistance in ohms. But in a rotor of a magneto, or for that matter any generator of electrical energy, there are other losses besides those ascribed to the resistance of the winding and the flow of the current. And it is the designer's problem to find the total of the losses, including the electrical losses in the windings, the hysteresis losses in the iron of the core, and if the conditions are such as to induce them, the losses due to Foucault current. The best way to measure these losses is to state them in watts per square inch

of the radiating surface of the armature, or other coil, as the case may be. As a fair statement if the peripheral velocity is say 50 feet per second, one-half watt per square inch of radiating surface may be permissible, and as a general proposition for changing conditions of temperature as measured in degrees centigrade, the following will hold:

$$W_2 = W_1 \times \frac{C_2^2}{C_1^2}$$

$W_2$  = watts lost per square inch at the higher temperature.

$W_1$  = watts lost per square inch at the initial temperature.

Restating these facts we may say:

$$C_2 = C_1 \times \frac{W_2}{W_1}$$

In these expressions we have evidence of the influence that an increased output has upon the temperature increase, and if account is taken of the fact that the radiating surface is constant in a driven coil, the conclusion may be reached that a burn-out is a logical deduction unless some means is afforded for the limiting of the watts output, if the speed of the rotor is increased, remembering that the electromotive force induced in the winding of a rotor (other things remaining constant) will increase in direct proportion to speed, it being the case that the voltage generated in a motor responds to the algebraic expression:

$$V_a = \frac{I_a \text{ sa } I_1}{10^8}$$

In which

$V_a$  = the voltage induced in the inductors on the rotor.

$I_a$  = the number of the inductors wound on the rotor.

$\text{sa}$  = the speed of the rotor in revolutions per second.

$I_1$  = the total of the magnetic flux emanating from one pole.

$10^8$  = the fundamental value of the volt, it being the case that one volt will be generated if 100,000,000 lines of magnetic force are cut per second by an inductor.

Analyzing this expression leads to the conclusion that the inductors, being a fixed quantity, cannot influence the situation by any manipulation of the instant, but if the speed of cutting of the lines of flux by the inductors is increased, as it will be if the speed of revolution of the rotor is increased, there must be a diminution of the total of the magnetic flux that emanates from one pole or the voltage will go up, compensating for the difference, and if it does, the current in amperes that will flow in the circuit will be increased according to Ohm's law, in which event the I<sup>2</sup>R losses will be augmented accordingly, and since the area of radiating surface is fixed, the watts lost per square inch will increase, and, as insulation is far from indestructible, the time will come when the magneto will burn out.

### The Threshold of the House of Profitable Investigation Is Ajar, Inviting the Receptive Mind to Enter and Partake of Its Just Reward.

Assuming that the discussion thus far has scratched the ground of trouble in the ignition system as it relates to a motor in service, there still remain the problems of the ignition system as they are confronted by the thermic conditions that obtain in the combustion chamber of the motor, in which the carbureter plays an important part. Here again the field is influenced by common practice, and the man who builds carbureters looks upon the effort as an abstract proposition, and the designer of the motor being busy with the type of trouble that he best understands, lets the carbureter man think for him in the most important part of his own field.

It is not the purpose here to make a demonstration in favor of the concentration of work, and it is even admitted that carbureters can best be made by specialists who do nothing else, but these advantages should not be handicapped by incoherence of relation, and it is suggested that common investigation under the leadership of the body politic of engineers, should so leave to the world knowledge along these lines, that the specialist in

the building of carbureters would know the needs of the specialist who is confined to the building of motors, all for the sake of harmony.

In the meantime, in order that we may appreciate the necessity for intelligent action, it would be almost enough to simply examine the variety of carbureters that are to be had for the asking, and to observe that they are almost indiscriminately used on the various types and sizes of motors, leaving but one conclusion, i. e., a large number of the applications must be dictated by commercialisms.

Even this view of the situation does not indicate that the carbureters are not good. For that matter, every one of them as made might outshine its neighboring example under certain conditions, but the point is that the promiscuous swapping of carbureters must be deemed as far from satisfactory.

In the further discussion of the carbureter problem, if a real gain is to be realized, it will surely be necessary to pay attention to the type of automobile gasoline that is to be mixed with the air in the carbureter. It cannot be said with even the possibility of accuracy that a mediocre design of carbureter will make an otherwise good motor perform up to the expectation of its designer, but it is only a step from a mediocre carbureter *per se* to mediocre carburetion brought about through the use of a good carbureter, and an inferior grade of the hydro-carbon liquid.

This discussion has placed the reader so that he is confronted by the greatest dilemma of the automobile art, making it necessary to say that the designing of the motor, fixing the compression, and the application of the carbureter can go no further in the direction of improvement until the Society of Automobile Engineers takes up the question of the standardization of the hydro-carbon liquids used for fuel, thus making it a feasible task to design the carbureter for the definite work that it will have to do, and to design the motor, fixing the compression, in view of the requirements, nor is it worth while to try to explain how any further good can come from any effort made, unless the designers of these units may be permitted to base their efforts upon a definite type of fuel, of which knowledge must be had of its normal characteristics, and the trend of its variables.

### Legislation May Be Necessary in the Long Run Fixing the Quality of Automobile Gasoline.

There is a superstition that has been bred upon the market, which has for its foundation the flimsy pretext that there is a famine of true gasoline, and based upon this superstition, dealers in automobile gasoline are reaping a harvest based upon perfidy, resulting in the use of everything from kerosene oil to water, so that the carbureters, good and bad, are laboring under an impossible handicap, and the motors that are designed with a compression that is satisfactory for true gasoline, has its compression value fixed too low for the composite fluid that may be had at 20 cents per gallon from half the garages in the land.

Until a committee has been assigned to the task of settling upon the desired characteristics of automobile gasoline, and that

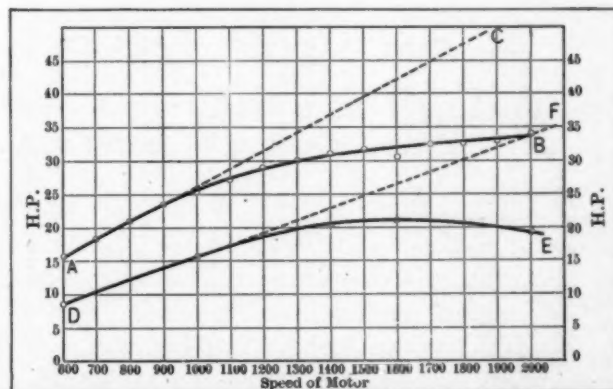


Fig. 6—Chart showing the throttling effect of a carbureter as applied to two different sizes of motors

committee has deliberated sufficiently to reach proper conclusions, it will not be possible for any man to tell the distillers of "crude" of the product that must be furnished to automobilists for their work, and it would be "unreasonable" to demand of these distillers that which cannot be described. But if a committee can be assigned to the work, and it deliberates sufficiently to arrive at a clear understanding of the needs, it will then be a feasible undertaking to go to the distillers of "crude," tell them what is wanted, and ask them to deliver a definite product.

It is a fair principle in law and in equity to demand of the vendor the definite thing that is ordered, and it follows, of course, that a "reasonable" price should be asked as compensation for the delivery of the definite goods desired. In view of the fact, however, that automobile gasoline is made up of one or more fractions of a series of the fractions of a distilling process, it is quite possible for the distiller to so manipulate the process that the actual composition of the product will be a variable over a wide range, but this is another way for saying that the distiller can deliver a definite product, and certainly the automobile business, if it is to make further progress, depends upon the use of a definite fuel.

#### Considering the Throttling Effect of Undersized or Complicated Types of Carbureters

If it is a fair principle in law and in equity that a buyer is entitled to a fixed measure of a definite thing if it is so ordered and payment is tendered, any violation of this principle, if it is the result of a chronic condition, launches the project into the realm of legislative enactment. As the situation stands to-day, the practice of delivering all sorts of mixtures of hydro-carbons to the users of automobiles on demand, can be no better described than to say it is on the border land of crime. At all events, it is a normal misdemeanor to deliver either short weight, or "watered milk," nor does the presence of "chalk" in the mixture justify the deception, and referring specifically to automobile gasoline, the practice of diluting the more volatile fractions with the fractions that can only be made to volatilize after they are swept into the combustion chamber, is on a par with the watering of milk, and it leads directly to the conclusion of the designer and it must ultimately bring about the downfall of the maker of the carbureter, who cannot hope to thrive and prosper if his device is a failure, as it surely will be if it is incapable of serving for the intended purpose. These are all matters that have direct bearing upon the horsepower rating of automobile types of internal combustion motors, it being the case that the horsepower output of a given motor depends:

- Upon the quality of the fuel used.
- Upon the quantity of the fuel that can be burned to carbonic acid and water within a given time.
- Upon the disposition of the relief heat units, taking account of those that deliver mechanical work.
- Upon the co-operative ability of the carbureter and the magneto.
- Mechanical efficiency of the motor.

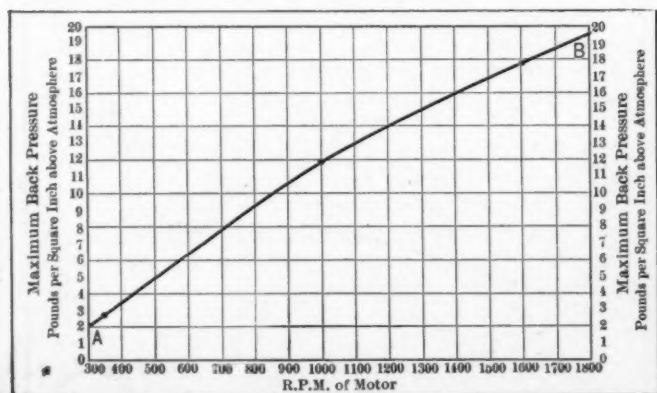


Fig. 7—Chart of the back pressure on a motor due to a muffler at varying speeds

Independent of the quality of the gasoline used, without losing sight of the importance of the same, the discussion is brought up to the question of the effect of a restricted intake, or of the obstruction offered by valve mechanisms or tortuous passageways, remembering that the mixture is drawn into the motor by suction due to the displacement of the piston and that the actual weight of the mixture decreases enormously as the depression in the intake increases as the latter is measured in pounds per square inch below the pressure of the atmosphere. The power of a motor varies enormously as the depression changes, and the conventional way of getting at the facts attending this situation is to find the value of the exponent "N" under the several conditions, the conventional formula for the undertaking being as follows:

Assume that  $B_1$  represents the piston displacement plus the combustion chamber space; in other words, the cylinder volume.

$p_1$  is the absolute pressure corresponding to the value  $V_1$ .

$p_2$  represents the absolute pressure corresponding to  $V_2$ .  
When,

$$p V^n = \left( \frac{V_1}{V_2} \right)^n$$

And,

$$n = \frac{\log p_1 - \log p_2}{\log V_2 - \log V_1}$$

In actual practice the values of the exponent "N" which is given as  $n$  in the formula above, ranges from 1.195 when the depression is 1.7 pounds per square inch below the atmosphere to as high as 1.40, this value having been found in a certain case when the depression was 2.7 pounds per square inch below the atmosphere. As a general proposition, the difference in performance of a motor from the power point of view may be as much as 20 per cent., if the exponent  $N$  changes between 1.3 and 1.25 in a given motor, and as fond as we are of describing the depression in our wares as a mere matter of a few sparse ounces, the fact remains that some of the motors have such a terrific depression that the carbureter wheezes so loudly that it disturbs the serenity of the owner of the car. These disconcerting noises that emanate from the region of the carbureter intake are as a loud voice telling the designer of the automobile that the power of the motor may be 20 or 30 per cent. below the amount that should be obtained from a motor of that size, especially if it lives up to the claims that are made for it.

But if a depression in the intake has so marked a bearing on the power of the motor it is almost nothing as compared with the strangling influence of a ruffian-like muffler that is placed to hide the fact that the designer of the motor recognizes his incapacity, and he tries to legislate for silence instead of coping with the trying causes of noise.

A very good way to chart the values of "N" is shown in Fig. 5 of a Franklin Motor, which is offered here as a suggestion of one of the methods in vogue in the plants where it is deemed expedient to cope with the inner problems. In this connection reference may be had to Fig. 6, which is a curve of a carbureter test, and the line A B shows how the carbureter limited the upward swing of the power and the line A C shows how the carbureter should have performed. A second motor was tested, using the same carbureter, and the amount of power taken from the motor was materially less, producing the line D E, but even in this case, the carbureter should have permitted the motor to deliver power which would produce the line D F. In this chart it will be seen that the horsepower is given as abscissa, and the ordinates are plotted in terms of speed of the motor in revolutions per minute. This is merely one of a considerable number of curves which have been examined, and while a good many of them show very much better results, this particular example is not the very worst among them.

Referring to Fig. 7, which is a chart showing the back pressure that was induced by the presence of a muffler on a given motor, it will be seen that this back pressure increased from A

at 300 revolutions per minute to B at 1,800 revolutions per minute. Measuring the back pressure in pounds, it was two pounds at 300 revolutions per minute, and it was 19.6 pounds at 1,800 revolutions per minute. This would not look like a serious matter to the casual observer, and yet to the engineer, any serious increment of back pressure is bound to annoy him beyond measure if he pauses for a moment and considers the fact that the power of the motor is proportional to the mean effective pressure, and that this mean effective pressure is in the neighborhood of 70 pounds per square inch in the average motor, and it is a very fine motor indeed that reaches the level of 80 pounds per square inch, and coupling these figures with the fact that one pound of back pressure offsets slightly more than one more pound of mean effective pressure, and it will be seen instantly that a throttling muffler is not to be tolerated for a moment. The thought occurs that a committee would have a fine time establishing the law of the type of muffler that induces silence without strangling the power functions of the equipment.

#### Considering the Equipment Employed for Cooling Purposes as It Influences the Power of the Motor

Speculation is at the helm in many of the efforts that are directed at the cooling problem. Point of view is rarely ever given the consideration to which it is due. No available source of information tells the extent of the trouble that follows in the footsteps of spheroidal action, and looking at the motors that are to be seen at every hand is sufficient to inform the observer that if there is anything in this phenomenon described as spheroidal action, the means for its cure are not at hand, and if the user of an automobile were to assume that discretion is the better part of valor, thus leading him to the thought that he should scrape the scale off the exterior surfaces of the dome of the cylinders, he would have a major general's job on his hands, in his attempt to find out how to gain access to the offending dome surfaces.

In the meantime the literature of fifty years tells every reader that 1-16 inch of depth of scale on a surface takes 37 odd per cent. from the ability of that surface to remit heat—in other words, to permit of the transfer of heat—and one of two things must be true; either designers are too liberal in their allowance of the ability of cooling water, or when 37 odd per cent. of the gateway of the heat is plugged up by scale, the section of the metal in the domes of the cylinder walls must then overheat and if it does it brings us back to the starting point and we are confronted by the necessity of doing something in a practical way to permit of getting at the surfaces and cleaning them off if the occasion requires. A matter of 20 years ago the writer was sent for to attack an unruly stationary gasoline motor that was set up in a little house on the Lake Shore north of Chicago, and it being in the Winter time the thought occurred

that the intense cold of that particular day had a lot to do with the mule-like qualities of the motor. After working with great persistence for several hours without success the odor of the "colored man" penetrated the distance from his hiding place in the wood-pile, and a further examination of the motor showed that a deposit had been precipitated out of the cooling water and that it had grown until the entire space within the water-jacket was filled up. The motor had been operating under these conditions for some little time perhaps and when the writer tackled the job all that was left was the semblance of a motor and 4,000 pounds of junk.

It is not the purpose here to say that automobile motors are troubled to the extent as indicated in the above example. Moreover, the conditions under which automobile motors operate are much more favorable, due to the fact that the water is used over and over, and the amount of deposit contained therein is insufficient to bring about a clogging up of the circulatory system. But if the radiator is not big enough for the work it is placed to do, or if the water pump fails in its duty and the water boils away the automobilist will have to replenish the water at frequent intervals, and in so doing he will afford the source of supply of scale, and it will settle down on the hottest surface, which is the dome of the cylinder walls, and it will there form a crust and spheroidal action will come, after which an overheated motor is the normal expectation.

From what has been said, and the subject is by no means exhausted, it would appear that a simple formula intended merely to satisfy the cravings of the disinterested individual is scarcely worth while. It is even a question as to whether or not any formula or set of formulæ can be sufficiently comprehensive to deliver the absolute facts in concrete cases. Of one thing we may be sure and that is there is a wonderful opportunity for a set of capable men to exercise their learned faculties, but in order to do so they will have to be provided with a laboratory, but it will necessarily have to be equipped with a wide variety of suitable apparatus; moreover, the time that the laboratory will have to devote to the intricacies of the motor problem will be far beyond the staying power of any single company.

#### Institute of Automobile Engineers (English) Have Made Considerable Headway in the Contriving of Formulæ for the Fixing of the Horsepower of Automobile Types of Motors

While this report was abstracted in THE AUTOMOBILE directly after it was made, it is believed that it will be opportune to present the report in full accompanied by the tables which were not spaced before, and particular attention is called to the enormity of the undertaking, and to the mass of data that resulted from the work done by the learned committee to which the task was assigned.

## The Rating of Petrol Engines

### Report of the Horsepower Formula Committee of the Institution of Automobile Engineers (English)

The committee consisted of Mr. Dugald Clerk, F. R. S., M. Inst. C. E. (chairman); Messrs. A. Craig, J. S. Critchley, C. R. Garrard, L. H. Hounsfield, Max R. Lawrence, Mervyn O'Gorman,\* L. H. Pomeroy and D. J. Smith (representing the Institution of Automobile Engineers); Colonel H. C. L. Holden, Captain R. K. Bagnall Wild, Professor Gallender, Dr. W. Watson, Messrs. W. Worby Beaumont and E. Russell Clarke (representing the R. A. C.), and Mr. G. A. Burls, M. Inst. C. E. (representing the S. M. & T.).

The report of the rating committee of the Institution was read and discussed at a meeting held on November 10, 1908. It dealt with pro-

posals put forward on behalf of the Society of Motor Manufacturers and Traders, by Mr. G. A. Burls, M. Inst. C. E. The rating committee agreed with the proposals in principle, but considered that the tests submitted did not support the modifications of the R. A. C. rating required by the formulæ. The report recommended the formation of a committee composed of members of the Institution, members of the R. A. C. and representatives of the Society of Motor Manufacturers and Traders. This committee was called "The Horsepower Formula Committee." Several meetings were held, and a scheme was agreed upon, Mr. G. A. Burls being good enough to undertake the collection of the necessary material from the leading automobile firms in this country and abroad; he has now prepared the tables of

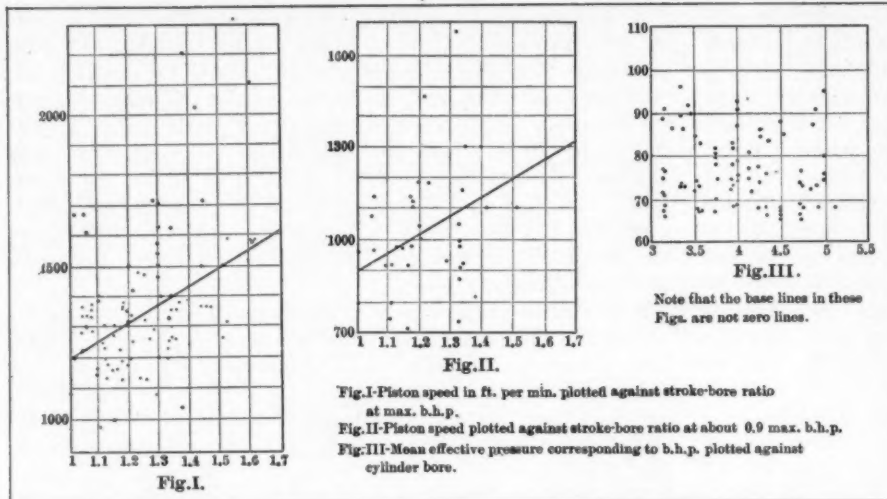
particulars which accompany this report, and made the many calculations and deductions shown. Mr. Burls' letter to the chairman of the committee accompanies the report, and it describes the nature of his work. At meetings held on the 30th of June and the 13th of October, 1909, the committee resolved to recommend for consideration of the following formula:

$$K d (ad + s) N$$

as a formula giving a rating with a stroke-bore correction. The constants K and s were purposely left without any specified value to enable the committee ultimately to arrive at values from experiment and observation. In this formula, d is the bore and s the stroke, in inches; N is the number of cylinders.

Before arriving at the above formula the com-

\*Mr. O'Gorman also represented the R.A.C.



mittee considered the two main corrections proposed to be applied to the R. A. C. formula, viz., variation of mean pressure with dimensions of cylinder, and variation of piston speed with ratio of stroke to bore. The R. A. C. formula corresponds to a mean effective pressure of 67.2 lb. per square inch per explosion, with a normal piston speed of 1000 ft. per minute. According to the R. A. C. formula, the rating =  $0.4 D^2 \times N$ . This simple formula, notwithstanding frequent statements to the contrary, involves both specified mean pressure and piston speed. It assumed, however, that piston speeds did not vary materially from 1,000 feet per minute. Examination of the accompanying tables of results proves conclusively that piston speed does increase with the stroke-bore ratio.

The tables give some particulars from tests made of 144 engines, but only 101 tests contain all the data required for comparing the effect of stroke-bore ratio on piston speed for values of  $r$  from 1 to 1.61, and they do not necessarily represent the maximum brake horsepower which can be obtained. The following table shows broadly the increase of piston speed with stroke-bore ratio. Five groups have been taken with a variation in each of 0.1 in stroke-bore ratio as nearly as could be obtained.

TABLE I.—CHANGE OF PISTON SPEED WITH VARIATION OF STROKE-BORE RATIO

Number of Tests	Ratio $r = s/d$	Highest recorded b.h.p.	Piston Speed at Max. B. H. P. ft. Per Min.
15	1.00 to 1.08		1,303
30	1.10 to 1.20		1,240
24	1.21 to 1.30		1,385
25	1.33 to 1.44		1,414
7	1.50 to 1.61		1,597
101			

Each value of piston speed is the mean of the values obtained in the number of tests given; to some extent, the taking of average results masks the variation between the different engines. Accordingly, Fig. I, has been prepared, in which piston speeds are plotted against stroke-bore ratio for the 101 engines.

The results plotted in Fig. I, show an undoubted increase of piston speed with stroke-bore ratio, but the results obtained are exceedingly irregular. For example, between  $r=1$  and  $r=1.1$  the maximum piston speed in three cases lies between 1,600 and 1,700 feet per minute, but the very large number of tests show 1,400 down to about 1,000 feet per minute. Between  $r=1.1$  and  $r=1.2$  the maximum values appear to be 1,400 and 1,500 feet per minute in two separate tests, but the greater number lie below 1,300 feet per minute. Between  $r=1.2$  and  $r=1.3$  the maximum values are above 1,600 feet per minute but the greater number lie below 1,400 feet per minute. The highest value of all is found between

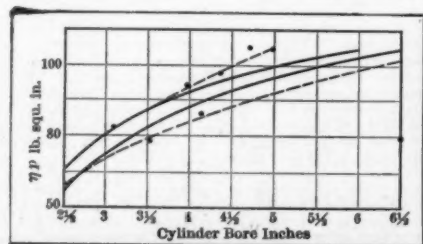


Fig. IV.—Showing increase of mean effective pressure with bore

$r=1.3$  and  $r=1.4$ , namely, 2,200 feet per minute—but it is understood that this figure is open to doubt. In the same way between  $r=1.4$  and  $r=1.5$  the highest value is just over 2,000 feet per minute, and the highest value at 1.6 is 2,100 feet per minute. From this it will be seen that it is impossible to formulate accurately any law of variation between stroke-bore ratio and piston speed. Notwithstanding this, if 1,200 feet per minute be taken as the speed to be expected with  $r=1$ , and 2,100 feet per minute that corresponding to  $r=2.5$ , then, assuming a linear law between the two points, the piston speeds found for different stroke-bore ratios are such as can be obtained (bench test) at maximum brake horsepower from a carefully designed engine. These piston speeds, it will be evident, are higher than those given by about 50 per cent. of the engines examined. About 50 per cent., however, give greater values, so that for a maximum rating no hardship would be occasioned by assuming the law to be that given by the black line shown upon Fig. I. Any formula required to express brake horsepower on this basis must assume a piston speed of 1,200 feet per minute for  $r=1$  and 2,100 feet per minute for  $r=2.5$ .

Forty of the engines, Nos. 1 to 50, of the appended tables were tested at approximately 0.9 of the highest brake horsepower recorded, and in

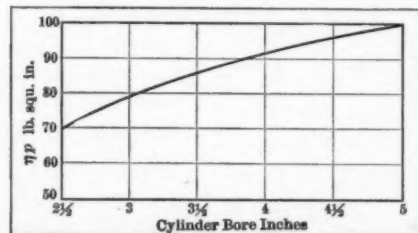


Fig. V.—Average relation between cylinder bore and mean effective pressure

all cases the tests show lower piston speed. These results have been plotted in Fig. II.

Here also the stroke-bore ratio increase is accompanied by increase in piston speed, but the piston speed for  $r=1$  may reasonably be taken at 900 feet per minute and for  $r=2.5$  at 1,800 feet per minute.

Mean Effective Pressure Corresponding to the Brake Horsepower.

This value has been called  $\eta p$  in the previous Reports. The results of the various tests have been tabulated and plotted against cylinder bore, but no increase can be deduced in this way as due to increasing cylinder bore. When, however, only one make of engine is taken the case is different, and the rise of  $\eta p$  with  $d$  is apparent.

Fig. III shows eighty-eight tests in which  $\eta p$  at maximum brake horsepower is plotted against cylinder diameter. The values considered are those given by tests from 124 engines. Out of these eighty-eight give mean effective pressures of 65 lb. per square inch and above. The pressures rise as high as 95½ lb. Expressed in percentages, approximately 5.6 per cent. of the 124 engines give values above 90 lb.; 15.3 per cent. between 80 and 90 lb.; 28.2 per cent. between 70 and 80 lb.; and 21.2 per cent. between 65 and 70 lb. About 50 per cent. of the engines thus show mean pressures above 70 lb. The highest pressure was obtained in a cylinder of 3.35 inches diameter. Fig. III, supplies no evidence of increase of mean pressure with bore. Some makers obviously succeed in getting very high mean pressures from quite small cylinders and others do not. It is interesting now to con-

sider the highest values of  $\eta p$  given in these tests by cylinders of different diameters. Compare the two highest obtained at the highest brake horsepower recorded with the cylinders of about 3 inches, 4 inches and 5 inches respectively as follows:

Test No.	$d =$	$\eta p$
122..	3.15 in.	90.5 lb. per sq. in.
83..	3.13 in.	88.
		Mean 89.2 lb. per sq. in.
45..	4.0 in.	92.3
88..	4.0 in.	90.2
		Mean 91.2 lb. per sq. in.
72..	5.0 in.	94.6
116..	4.88 in.	90.3
		Mean 92.4 lb. per sq. in.

Here an increase is shown of nearly 4 per cent. The value of  $\eta p$  for about 3 inches diameter is 89 lb. per square inch and for 5 inches diameter 92.5 lb. per square inch.

Tests by Messrs. White and Poppe cited by Mr. Burtis show a large increase—over 25 per cent.; 3.15 inches diameter giving  $\eta p=80.6$  lb. per square inch, and 5 inches diameter  $\eta p=102$  lb. per square inch. See Fig. IV.

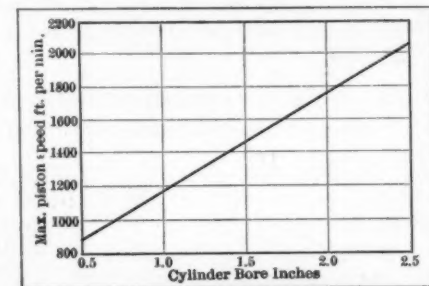


Fig. VI.—Maximum piston speed plotted against cylinder bore

The experience of many makers also proves that in testing under similar conditions increase of cylinder dimensions increases  $\eta p$ . In discussion many members of the committee give this increase as part of their ordinary experience. It is known also that in closed cylinder experiments the cooling loss in a given time is greater in a small vessel than in a large one; further, that in gas engine practice higher mean pressures are obtained under similar conditions of compression in large cylinders than in small; also that the mechanical efficiency of a large engine is usually greater than that of a small one. Taking into consideration all these matters and the results of many tests of petrol engines for cars, the committee is of opinion that it is necessary to allow for an increase of the value of  $\eta p$  with the cylinder diameter, and they accept 68½ lb. per square inch for  $\eta p$  when  $d=2\frac{1}{2}$  inches and 99½ when  $d=5$  inches. The best current practice as to  $\eta p$  is therefore taken as

$\eta p = 130 (1 - 1.18/d)$  lb. sq. in. (a)  
After consideration of the evidence contained in the data of 143 petrol engines collected by Mr. G. A. Burtis, M. Inst. C. E., the committee is of opinion that the rate of increase of maximum practicable piston speed with stroke-bore ratio can be adequately represented by the equation  
 $s = 600 (r + 1)$  ft. per min. (b)  
where  $r$  is the ratio of stroke to bore. This implies a maximum practicable piston speed of 1,200 feet per minute for  $r=1$ , rising to 2,100 feet per minute for  $r=2.5$ . See Fig. VI.

Proposed Formula.  
The committee therefore propose a formula which includes an increase of  $\eta p$  with cylinder diameter and increase of piston speed with stroke-bore ratio. It is:  
Max. b.h.p. rating per cylinder =  $0.464 (d + s) (d - 1.18)$

This formula may be considered to give the maximum practicable brake horsepower as determined by a bench test under onerous, but still safe, conditions for carefully designed and soundly constructed engines of from 2½-inch to

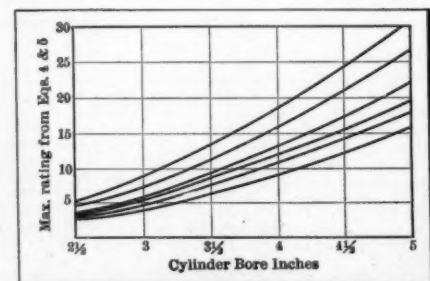


Fig. VII.—Maximum engine ratings plotted against cylinder bore

5-inch cylinder diameter and stroke-bore ratio up to 2.5.

The exact formula for horsepower in terms of bore,  $d$ , and piston speed,  $\sigma$ , is:

b.h.p. per cyl. =  $1/168,000 \cdot d^2 \cdot \eta_p \cdot \sigma$  . . .  
where  $d$  is in inches;  $\eta_p$  in pounds per square inch, and  $\sigma$  in feet per minute.

In this equation substitute for  $\eta_p$  and  $\sigma$  the expressions (a) and (b), and the equation then becomes:

$$\text{b.h.p. per cyl.} = 0.464d(d-1.18)(r+1).$$

For purposes of computation the following equivalent form is better:

$$\text{b.h.p. per cyl.} = 0.464(d+s)(d-1.18).$$

To simplify calculations, we recommend that the constant in the formula be taken as 0.45 instead of 0.464, that is the brake horsepower for an engine with  $N$  cylinders =  $0.45(d+s)(d-1.18)N$  where  $d$  and  $s$  are in inches. The form of the formula if  $d$  and  $s$  are in millimeters will be added later.

## Proposed Maximum Power Rating Formula

The following paper by G. A. Burls, Wh. Ex., M. Inst. C. E., is supplementary to the report above given, takes up the same problem in a more detailed manner, and includes tables of maximum piston speeds and revolutions, variation of compression ratio, maximum b.h.p. per cylinder and valve diameter and lift.

The following symbols are employed throughout:

- $d$  = bore of cylinder in inches.
- $s$  = stroke of piston in inches.
- $r = s/d$  = the stroke-bore ratio.
- $N$  = number of cylinders.
- $n$  = number of revolutions per minute.
- $\sigma$  = piston speed in feet per minute.
- $\eta$  = mechanical efficiency of engine.
- $p^1$  = "acceleration-pressure" in pounds per square inch on piston.
- $p$  = mean effective pressure on piston in pounds per square inch during the working stroke.
- $m$  = mass of piston and connecting rod, all complete, in pounds.
- $l$  = crank length in feet.
- $l$  = connecting rod length, between centers, in feet.
- $v$  = mean velocity of gas through inlets, in feet per minute.
- $\delta$  = diameter of inlet valve in inches.
- $\lambda$  = lift of inlet valve in inches.
- b.h.p. = brake horsepower.
- $C$  = volume ratio of compression.
- $P$  = compression pressure in pounds per square inch, absolute.

In the accompanying diagram (Fig. I.) values of the stroke-bore ratio are plotted against bore. No regular variation was, of course, to be anticipated, but the diagram indicates a steady general diminution in  $r$  with increase of  $d$ . Broadly, in the cases included in Fig. I.,  $r$  varies from 1.75 for  $d = 2\frac{1}{2}$  inches, to 1.0 when  $d = 6$  inches. Mr. Poppe holds the opinion that for large bores, from considerations of gearing strength, and comfort of occupants of the car, a value of  $r$  not differing much from unity is best; for small bores  $r$  may be  $1\frac{1}{2}$  or even more without much disadvantage; during the past two or three years the value of  $r$  has shown a tendency to increase. Few examples exist at present wherein the ratio has as high a value as 2; practice tending towards the use of as large a ratio as can be adopted without undue sacrifice of smoothness in running. For the car engines of 1911 the range appears to be from  $r = 2.0$  for  $2\frac{1}{2}$ -inch bore, to

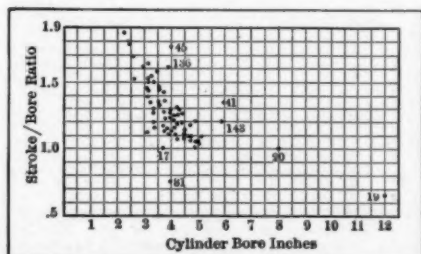


Fig. I.—Charting cylinder bore in its relation to the ratio of stroke to bore

TABLE OF PROPOSED RATINGS BY FORMULA.  
Max. b.h.p. rating per cylinder =  $0.45(d+s)(d-1.18)$ .

$r =$	$d =$ in.	$s =$ in.	per Cyl.	$r =$	$d =$ in.	$s =$ in.	per Cyl.
0.75	2 1/2	1 7/8	2.6	2 1/2	3 1/2	3.7	
	3	2 1/4	4.3	3	4 1/2	6.1	
	4	3	8.9	1.5	3 1/2	5 1/4	9.1
	5	3 3/4	15.0		4	6	12.7
					4 1/2	6 3/4	16.8
					5	7 1/2	21.4
1.0	2 1/2	2 1/2	3.0	2 1/2	5	4.5	
	3	3	4.9	3	6	7.4	
	3 1/2	3 1/2	7.3	2.0	4	8	15.2
	4	4	10.2		5	10	25.8
	4 1/2	4 1/2	13.5		5	10	25.8
	5	5	17.2		5	10	25.8
	2 1/2	3 1/2	3.3	2.5	3	6 1/2	5.2
	3	3 3/4	5.5		4	7 1/2	8.6
1.25	3 1/2	4 3/8	8.2		4	10	17.8
	4	5	11.4		5	12 1/2	29.6
	4 1/2	5 3/4	15.1				
	5	6 1/4	19.3				

$r = 1.0$ , or even rather less, for 6-inch bore; the short table at the top of this column illustrates generally the state of current practice.

The lowest value of  $r$  is found in the 20-horsepower Lanchester, with a bore of 4 inches and a stroke of only 3 inches; thus  $r$  is here only 0.75; in this respect the Lanchester engine is unique. A stroke of 3 inches also appears to be the shortest used in car engines; the longest, excluding racing engines, is about  $7\frac{3}{4}$  inches, though it is rare to find an engine with a stroke exceeding 7 inches. Pistons are usually of medium, hard, close-grained cast iron, usually with three, but occasionally with four cast iron

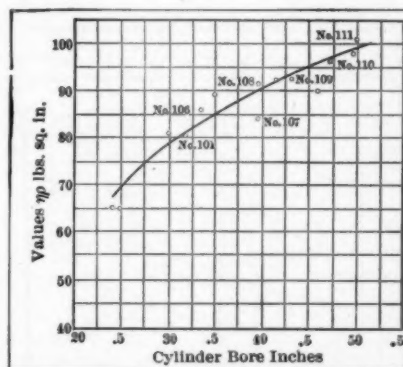


Fig. II.—Charting the variations of pressure in pounds per square inch to bore of cylinders

spring rings; an additional oil-excluding ring is sometimes fitted near the bottom of the piston. In normal car engines the piston is generally rather greater in length than the cylinder bore, the proportion averaging about 1.16 to 1.

TABLE ILLUSTRATING ROUGHLY THE VARIATION OF  $r$  WITH  $d$  IN CAR ENGINES OF 1911.

Description.	Nominal Bore Stroke			
	h.p.	m.m.	m.m.	Bore.
<b>Small car engines:</b>				
Calthorpe .....	15.0	75	150	2.0
Le Gui .....	10.0	65	130	2.0
Le Gui .....	15.0	75	150	2.0
Jackson .....	6.2	100	200	2.0
Delage .....	15.7	65	125	1.92
Opel .....	14.0	64	120	1.88
Delage .....	9.5	62	110	1.78
<b>Medium car engines:</b>				
Sunbeam .....	18-22	80	120	1.5
Talbot .....	20	80	120	1.5
Sheffield Simplex .....	25	85	127	1.49
Napier .....	30	88	127	1.55
Deasy .....	18-24	90	130	1.45
Crossley .....	20	102	140	1.37
<b>Large car engines:</b>				
Dennis .....	40	126	130	1.03
Lanchester .....	38	102	102	1.0
Maudslay .....	35-45	127	127	1.0
Napier .....	65	127	127	1.0
Talbot .....	35	127	120	0.945
Imperial .....	56-60	150	140	0.933
Napier .....	90	156	127	0.814

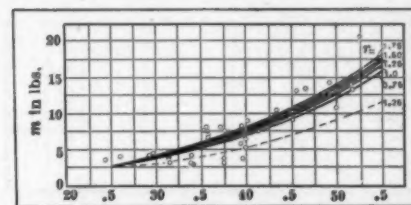


Fig. III.—Weight of reciprocating parts in terms of bore and ratio  $r$ .

Pressed steel pistons machined all over are now being increasingly used, as with this material from 15 to 20 per cent. of weight can be saved.

Connecting rods are of stamped steel, of I section; the ratio of length of rod to stroke of piston varies but little at present from the value  $9/4$ .

In the S. M. M. T. report on horsepower formulae no actual values of  $m$  are given, but it is assumed from rational considerations alone that  $m$  can be expressed in the form:

$$m = ad^2(d+bs)$$

$$m = ad^3(1+br),$$

where  $a$  and  $b$  are constants.

In the tables of engine data we have now the advantage of possessing a large collection of actual values of  $m$  over a range of engine sizes from  $d = 2.44$  inches to  $d = 12$  inches.

The author has found it necessary to add a constant term to the above formula in order that it may adequately resume practice, and suggests the following expression:

$$m = 0.08d^3(1+0.15r) + 1.5 \text{ lb.} \quad (1)$$

for cast-iron pistons. For the very light pressed steel pistons now being used the few cases collected suggest the expression

$$m = 0.05d^3(1+0.15r) + 1.5 \text{ lb.} \quad (2)$$

In the accompanying Table II. values of  $m$  and of  $d^2s/m$  and  $\sqrt{d^2s/m}$  are shown, calculated from equation (1) for the whole range of present practice; and in Fig. III. the graphs of (1) and (2) are exhibited for several values of  $r$  together with plottings of actual values of  $m$  taken from the tables of engine data.

In Tables III. and IIIA. actual and calculated values of  $m$  may be compared; it will be noted that equation (1) fairly resumes the facts from  $d = 2\frac{1}{2}$  inches to  $d = 12$  inches.

In Fig. IV. values of  $d^2s/m$  from Table II. are plotted against  $d^2s$  for values of  $r$ , from 0.75 to 2.5 inclusive; the curves are hyperbolae, having horizontal asymptotes as indicated in the diagram. The asymptotes are rapidly approached, but unfortunately  $d^2s/m$  does not become approximately constant in value until  $d^2s$  exceeds 125, which is somewhat outside the range of ordinary sizes. Much attention is now being given to the reduction of the mass of the reciprocating parts, not only in order to lessen vibration arising from want of complete balance, but also to enable higher piston speeds to be normally used, and thus diminish the weight of the engine per horsepower. At the very high piston speeds now frequent in car engines the inertia of the piston and connecting rod exercises an important influence upon the distribution of the driving effort of the crank pin during the working stroke. In the accompanying Fig. V. OP represents the crank, PM the connecting rod of an engine; the piston stroke is AB; DOCE is the curve of piston acceleration obtained by any of the well-known graphical methods, for example, that of Prof. Klein. In the position shown MQ measures the acceleration of the piston to the scale for which OP measures the velocity of the crank pin.

The values of the acceleration when the piston is respectively at the top and bottom of its stroke are:

$$\text{top} \quad AD = \omega^2 p (1 + p/l) \text{ ft. per sec. per sec.} \quad (3)$$

$$\text{bottom} \quad BE = \omega^2 p (1 - p/l) \text{ ft. per sec. per sec.} \quad (4)$$

The force necessary to produce this acceleration

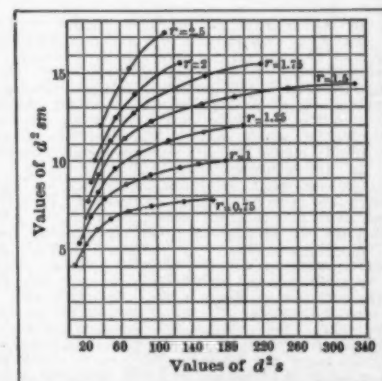


Fig. IV.—Chart showing the values of  $d^2 s/m$  in relation to the values  $d^2 s$



seen that there is no observable tendency to any diminution in value with increase in bore from 2½ inches to 6 inches.

If  $C$  denote the compression ratio, and  $P$  the compression pressure in pounds per square inch absolute, then for practical purposes

$$P = 14.7 C^{\frac{4}{3}} \quad (14)$$

The maximum compression pressure that can be successfully used in petrol car engines at present appears to be about 135 lb. per square inch absolute, corresponding to a compression ratio of about 5½. Much difference of opinion has existed on the point of the variation of  $\eta$  with size of engine, and it may accordingly be well here to review some of the evidence available in support of the view that an increase should be regarded as taking place with increase of bore.

Mr. Remington says: "By adopting suitable proportions of carbureter, induction and exhaust pipes and valves, it is possible to fill and scavenge an engine running at very high piston speeds and thus obtain as high a mean effective pressure as is usually associated with moderate, or even low, piston speeds." And again: "From an examination of the normal tests of the Wolseley engines many lessons have been learned, conspicuous among them being the increase of brake mean effective pressure with increase of size."

Prof. Callendar, from the results of some tests over a limited range, was led to propose the relation:

$$\eta = k (1 - 1/d) \quad (15)$$

as expressing the connection of  $\eta$  with  $d$ .—(Proc. I.A.E., 1906-7.)

The Power Rating Sub-Committee of the Society of Motor Manufacturers, after consideration of a number of further cases, adopted an expression of this form in their report on horsepower formula issued in August, 1908. In a subsequent discussion of this report by the Institution of Automobile Engineers, Prof. Callendar said: "The experiments of the A. C. F. on 96 engines ranging very equally from 65 mm. to 190 mm. (i.e., 2½ to 7½ inches) in bore . . . are practically conclusive evidence of the increase of mean pressure with size." And further: "We cannot be far wrong if we take the formula  $k d$  ( $d-1$ ), which allows only half the rate of increase observed by the A. C. F. . . . to represent the increase of  $\eta$  with  $d$ ."

Mr. P. A. Poppe says: "We have now tested over 1,700 engines of bore from 3.15 inches to 5 inches, the formula we have arrived at for the calculation of brake horsepower at a piston speed of about 1,000 feet per minute is:"

$$b.h.p. = 0.81 (d - 0.79)^{\frac{3}{2}} \quad (16)$$

Comparing this with equation (13), we see that its form implies that

$$\eta = 136 (1 - 0.79/d)^{\frac{3}{2}} \quad (17)$$

so that in these tests an  $\eta$ -increase with  $d$  is manifested. The following are results from equation (17):

$d = 2\frac{1}{2}$ in.	3 in.	4 in.	5 in.	6 in.
$\eta = 63\frac{1}{2}$	74	87	96	102½

TABLE ILLUSTRATING THE IRREGULAR VARIATION OF COMPRESSION RATIO IN PRACTICE.  
1910 Car Engines.

Bore in.	Stroke in.	Volume ratio of compression.	Compression pressure lb. per sq. in. abs.
2.44	4.33	4.1	96
2.56	4.33	4.5	109
2.60	3.94	4.5	109
2.95	4.73	4.5	109
3.13	4.75	3.1	67
3.15	4.73	4.0	93
3.38	4.0	4.8	119
3.50	4.5	4.7	115
3.54	4.73	3.23	70
3.54	4.73	4.8	119
3.56	5.12	5.1	130
3.74	5.32	4.0	93
4.0	4.92	5.0	126
4.0	5.5	3.0	64
4.0	3.0	4.2	99
4.13	5.0	4.9	123
4.5	5.0	3.78	87
4.75	5.0	4.7	115
4.88	5.12	4.8	119
4.92	5.92	3.8	88
5.0	5.25	5.1	130
5.12	5.52	4.25	101
5.92	7.1	4.2	99

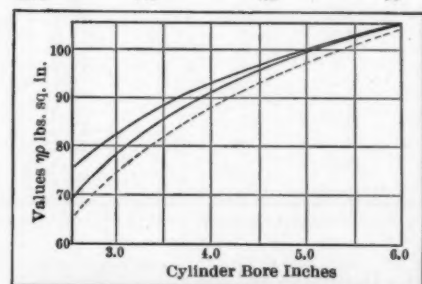


Fig. IX.—Abscissa in terms of pressure and results obtained involving the bore of the cylinder in inches

The last table above shows the results obtained from tests of a series of engines built by the same firm; it will be noted that the value of  $\eta$  steadily rises as the engines increase in size. The piston speed in each case was about 1,000 feet per minute; plotting the values of  $\eta$  against  $d$ , we get the curve shown by the upper line in Fig. IX. The lower dotted line shows the variation in accordance with equation (17), representing Mr. Poppe's experience.

From the tables of engine data, Table I has been constructed, including 67 engines arranged in order of increasing bore, at a piston speed in the neighborhood of 1,000 feet per minute. The volume ratio of compression is also tabulated in order to emphasize the absence of any observable connection with  $\eta$ ; this is probably due in many cases to inadequate valve and piping areas and is also largely influenced by the valve setting. In Table I we are clearly entitled to reject low values of  $\eta$  in any group. After consideration, the writer has thought it fair, on the whole, to take the average of the three highest figures in each set; the result is shown in Fig. II, where the points are plotted down on the diagram, in black dots.

Excepting the anomalous result for  $d = 4\frac{1}{2}$  to

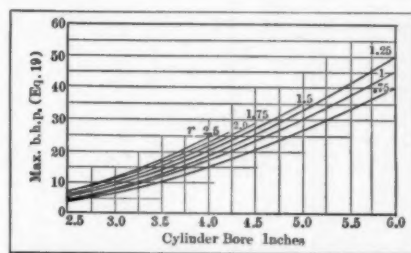


Fig. X.—Curves plotted from data involving the bore of the cylinder in inches and maximum power as found by equation 19

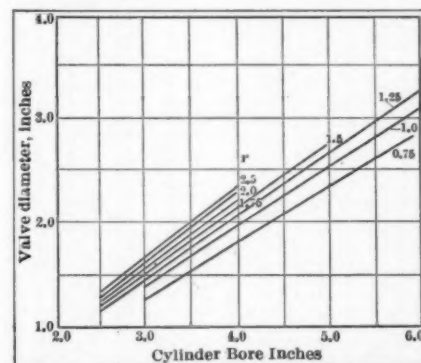


Fig. XI.—Charting on a basis of valve diameter in inches in relation to the bore of the cylinder in inches

$4\frac{1}{2}$  in., a decided increase of  $\eta$  with  $d$  is manifested, and on trial it is found that the formula  $\eta = 130 (1 - 1/0.85d)$  . . . (18) best resumes the points, as is indicated by the dotted curve.

The following values of  $\eta$  are furnished by equation (18):

$d = 2\frac{1}{2}$ in.	3 in.	4 in.	5 in.	6 in.
$\eta = 69$	79	91½	99½	104

These are plotted down on Fig. IX and furnish the middle curve shown by a full black line. The three curves are in substantial agreement; the writer herein adopts the mean, viz., equation (18), to express the variation of  $\eta$  with bore. Taking, then, equations (10) and (18) and substituting for  $\sigma$  and  $\eta$  in equation (13), we obtain the following rating expression for maximum performance:

$$\text{Max. b.h.p. per cyl.} = \frac{1}{2} d (d - 1.18) \sqrt{d^2 s / m} \quad (19)$$

The following table, calculated from this equation, gives figures for the range of sizes usual in car engine practice, the values of  $\sqrt{d^2 s / m}$  being taken from the table. In Fig. X also curves are shown connecting max. b.h.p. per cylinder with bore, in accordance with equation

TABLE SHOWING INCREASE OF  $\eta$  WITH  $d$  ON TEST OF A SERIES OF ENGINES BUILT BY THE SAME FIRM.

$d$ in.	$s$ in.	$n$ r.p.m.	B.H.P. by test.	$\eta$ lb. per sq. in.
3.15	3.54	1675	18.7	80.6
3.35	4.33	1360	22.6	86.2
3.94	4.33	1360	32.0	88.2
3.94	5.12	1150	33.3	91.8
4.33	5.12	1150	40.8	93.0
4.73	5.12	1150	50.6	96.5
5.0	5.12	1150	59.5	102

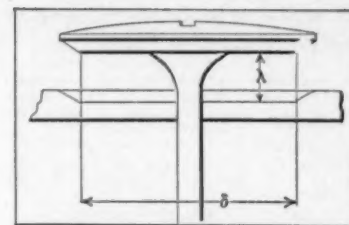


Fig. XII.—Diagrammatic presentation of a valve and the seat for use in illuminating the text

(19). The results furnished by equation (19) are very high, and can only be approximated to at present for short periods of running and under the special conditions of racing where every refinement of practice is utilized. They appear, however, to be not unreasonable when compared with some actual racing engine performances, as the following six cases show:

(1.) A four-cylinder 16-20 h.p. Sunbeam engine developed on test 54.2 b.h.p. at 2,300 revs. per minute; i.e., 13.55 h.p. per cylinder. Here  $d = 3.74$  in.,  $s = 5.32$  in., and  $m = 7.1$  lb., whence, by equation (19):

$$\text{Max. b.h.p. per cyl.} = 15.5.$$

Thus in this case the actual power was 87 per cent. of the rating according to equation (19).

(2.) The 100 mm. X 250 mm. (3.94 in. X 9.85 in.) single cylinder de Dion racing engine was referred to in the Autocar for May 14, 1910, as follows: "It is claimed that this engine will develop about 35 horse-power, but it is probable that 30 will be nearer the mark."

Here:

$$d = 3.94 \text{ in. } s = 9.85 \text{ in. } r = 2.5 \quad m = 5.7 \quad \sqrt{d^2 s / m} = 5.18$$

whence  $\sigma$  max. = 3,420 ft.p.m.,  $n$  max. = 2,080 r.p.m., and from equation (19):

$$\text{max. b.h.p.} = 28,$$

which is in satisfactory agreement with the anticipated performance. This engine was fitted to the winning Le Gui car in the Coupe des Voiturettes competition of June, 1909; the course of roundly 280 miles was covered at an average speed of 47½ miles per hour.

(3.) The 1908 Coupe des Voiturettes competition was won by a Delage car, fitted with a single-cylinder engine of 3.94 in. bore and 5.92 in. stroke. Using equation (2) we have  $m = 5\frac{1}{2}$  lb.; the actual mass is said to have been very small but the author does not know the exact figure. With  $m = 5\frac{1}{2}$  we have  $\sqrt{d^2 s / m} = 4.2$ , whence, by equation (19):

$$\text{max. b.h.p.} = 22.8.$$

The author is informed that the actual power developed exceeded this figure; this may well have been the case if the value of  $m$  were very low. The car covered the whole course at an average speed of just 50 m.p.h., so that the average horsepower, roughly estimated from the gross weight of car, and wind resistance, must have exceeded 20.

(4.) In the Autocar for December 17th, 1910, an illustration appears of a four-cylinder 130 mm. X 190 mm. Fiat engine nominally of 90 horsepower, and it is stated that this engine gives 143 horsepower on the brake, i.e., 35.8 b.h.p. per cylinder.

Here  $d = 5.12$  in.,  $s = 7.48$  in.,  $\sqrt{d^2 s / m} = 3.66$ ; substituting in equation (19), we have therefore

$$\text{max. b.h.p. per cyl.} = 36.8,$$

a result practically identical with the stated maximum output.

(5.) In the Autocar for October 29th, 1910, there appeared an illustration and note on the performance of a 4-cylinder White and Poppe engine, 80 mm. X 130 mm. (3.15 in. X 5.12 in.), having an R.A.C. rating of 15.9. An "all out"

MAXIMUM B.H.P. PER CYLINDER FROM EQ. (19).

	Cast Iron Pistons.			Max. b.h.p.
	Max.	Max.		
0.75	2.5	3.33	2.5	4.6
	3.0	6.22	3.0	8.6
	3.5	10.0	3.5	13.6
	4.0	14.5	4.0	19.8
	4.5	20	4.5	27.0
	5.0	26	5.0	35.2
1.0	2.5	3.8	2.5	4.9
	3.0	7.0	3.0	9.2
	3.5	11.5	3.5	14.7
	4.0	16.5	4.0	21.2
	4.5	22.7	4.5	28.8
	5.0	29.5	5.0	37.6
1.25	2.5	4.2	2.5	5.7
	3.0	7.9	3.0	10.6
	3.5	12.5	3.5	17.0
	4.0	18.3	4.0	24.4
	4.5	24.9		
	5.0	32.7		
	5.5	41.2		
	6.0	50.6		

test of this engine furnished the following results:—

r.p.m. ... 1900 2000 2200  
b.h.p. ... 38 44 47

Thus the maximum horse-power per cylinder was 11.75. The actual mass,  $m$ , of the reciprocating parts is not stated; if this be estimated by aid of equations (1) and (2) we get:—

For steel pistons ...  $m=3.44$  lb.

Hence  $\sqrt{d^3 s/m}$  has in this case the value 3.84. Using equation (19) we have therefore for the maximum power rating:—

Max. b.h.p. per cyl. 11.9 with steel pistons.

A result again in entire accord with the actual performance.

(6.) The 4-cyl. 3.54 in.  $\times$  4.73 in. Vauxhall engine in 1910 developed about 52 b.h.p. at 2,400 r.p.m. Writing in January, 1911, Mr. Pomeroy states:—"We have now got this up to 60 b.h.p. at 2,500 r.p.m., mainly by reduction of reciprocating weight and improvement in mechanical efficiency." Here by the proposed maximum rating formula, using  $m$  for cast-iron pistons, we have:—

Max. b.h.p. per cyl.=13.5,

while with  $m$  for steel pistons:—

Max. b.h.p. per cyl.=15.8.

The actual maximum powers being 13.0 and 15.0 respectively; the correspondence is here again very close.

From these instances it is clear that under racing conditions the ratings furnished by equation (19) are in satisfactory agreement with actual performances, and the writer therefore proposes the use of a formula of this form for rating purposes in "all out" contests for the future. To obtain the rating of any particular engine by the aid of this it would be, strictly considered, necessary to actually weigh the reciprocating parts in order to determine  $m$ . If this be considered impracticable,  $m$  could be inferred with a sufficient degree of accuracy from a formula of the type of equation (1) or (2) as soon as the bore and stroke were known. The great discrepancies in the power results frequently observed in engines of about the same size are largely due to the want of uniformity in practice in regard to valve diameter and lift, piping and valve setting. Some makers purposely fit small valves to prevent overrunning of the engines; even with the same engine builder there is, however, frequently found to be considerable want of uniformity in valve proportions, considerations of economy in production sometimes resulting in the same size or lift of valves, or both, being used in several engines of different bore and stroke, as the following series of figures illustrates. The upper line shows the actual valve diameters, the engines increasing in size from left to right; the lower line gives the calculated diameters from an equation to be given shortly:—

Actual  $\delta$  :— 1.16 1.58 1.58 1.61 1.61 1.61 2.36

Calculated  $\delta$  :— 1.31 1.40 1.63 1.63 1.80 1.97 2.08

During the suction stroke  $\pi/4 \cdot d^3 \cdot s$  cubic inches of mixture enter the cylinder through the inlet valve. If the engine makes  $n$  revs. per minute, there are on the Otto cycle,  $n/2$  suction strokes per minute, and hence  $n/2 \cdot \pi/4 \cdot d^3 \cdot s$  cubic inches of mixture must pass the inlet valve per minute. If  $v$  denote the average velocity through the valve area  $\pi/4 \delta^2$ , in inches per minute, we have therefore:—

$$\pi/4 \delta^2 v = n/2 \cdot \pi/4 \cdot d^3 \cdot s$$

$$v = 3/4 \cdot n \cdot d^3 \cdot s / \delta^2$$

$$v = 3/4 \cdot \pi \cdot d^3 \cdot s$$

whence, if  $v$  denotes the mean velocity in feet per minute, as  $v = v'/12$ , we get:—

$$v = 3/4 \cdot (\delta/d)^2 \cdot s \cdot \sigma \quad (20)$$

where  $\sigma$  is the piston speed in feet per minute.

Cylinder Bore Inches.

Values of  $v$ , calculated in this way are given in the tables of engine data herewith, and consideration of the figures thus obtained has led the writer to propose that an average value of  $v$  be taken as 2000\* feet per minute.

For attainment of the best power result, this should be reached at the maximum piston speed; substituted, then in (20) 2,000 for  $v$ , and for  $\sigma$  the value given in equation (10), and we get on reduction:—

$$\delta = 0.287 \cdot d \cdot \sqrt{d^3 s/m} \text{ in.} \quad (21)$$

This equation connects the valve diameter with the cylinder bore, and the function  $d^3 s/m$  on the basis of maximum power rating herein proposed. Values of  $\delta$  calculated from the equation are shown in the tables of engine data, and also in the accompanying table and the curves in Fig. XI.

Reference to the table of engine data shows the great want of uniformity existing in current practice. In general, the values of  $\delta$  furnished by equation (21), are somewhat in excess of actual sizes, though cases will be noted in which engines have valves fully equal to the calculated diameter. If  $\lambda$  be the lift of the valve, Fig. XII., a cylinder surface  $\pi \delta \lambda$  is available for the passage of gas normally across it; if we equate this to the throat area,  $\pi/4 \delta^2$ , we have, on reduction:—

$$\lambda = \frac{1}{4} \delta$$

as the valve lift corresponding to diameter  $\delta$ . So great a lift as one quarter of the diameter is, however, unusual in car engines on account of inertia; at very high speeds the tappet rod parts contact with the cam even when the strongest springs are employed. To overcome this difficulty and obtain sufficient valve area, some racing en-

TABLES OF ENGINE DATA REFERRED TO IN

No. of Eng.	Bore in. =d.	Str. in. =s.	Str. bore Ratio =s/d	No. of Cylinders. N.	Vol. Ratio of Compression.	INLET AND EXHAUST VALVES.		Mean Velocity through Inlets at Maximum H.P.=v, ft. p.m.	Value of $\delta$ given by $\frac{.29d\sqrt{d^3 s}}{m}$ inches.	Mass of Reciprocating Parts in lb. =m.	$d^3 s=$	$\sqrt{d^3 s}$	$\eta$ at Max. B.H.P. lb. per sq. in.
						Diameter and Lift of Inlets. $\delta$ in. $\times$ lin.	Diameter and Lift of Exhausts. $\delta$ in. $\times$ lin.						
1	3.94	5.12	1.30	4	4.42	1.97 $\times$ .296	1.97 $\times$ .296	1700	2.04	7.63	79.3	3.22	69.3
2	3.50	4.02	1.15	4							49.3		59.3
3a	4.92	5.0	1.02	6							120.8		72.6
4	4.73	5.0	1.06	4	4.73	2.0 $\times$ .31	2.0 $\times$ .41	> 1870	2.39	12.22	112.0	3.03	66.3
5	4.13	5.0	1.21	4	4.9	1.75 $\times$ .22	1.75 $\times$ .31	> 1980	2.17	7.88	85.3	3.29	57.3
6b	3.74	5.32	1.42	4	4.0	1.97 $\times$ .315	1.97 $\times$ .315	1840	1.96	7.00	74.3	3.26	79.6
7c	3.78	5.12	1.35	4							73.2		
8c	4.88	5.12	1.05	4							122		
9	3.35	4.33	1.29	4							48.7		72.7
10	4.0	4.53	1.13	4							72.5		75.2
11	4.17	4.57	1.10	4							79.2		71.2
12	4.73	5.12	1.08	4							114.7		73.3
13	4.88	5.12	1.05	4							122		86.7
14a	4.73	5.00	1.06	6							112		68.2
15	4.25	5.00	1.18	4	3.52	1.75 $\times$ .375	1.75 $\times$ .375	1450	2.26	8.06	90.3	3.35	77.3
16	4.5	6.00	1.33	4							121.5		59.0
17	3.75	3.75	1.0	4	3.78	1.875 $\times$ .31	1.875 $\times$ .31	> 975	1.77	7.56	52.8	2.65	69.8
18	4.5	5.0	1.11	6							101.2		66.7
19d	12.0	8.0	0.67	4	3.18	5.0 $\times$ .81	5.0 $\times$ .81	> 1375	5.78	151	1152	2.76	62.5
20d	8.0	8.0	1.0	4	3.07	3.38 $\times$ .88	3.38 $\times$ .88	> 1850	4.08	54.0	512	3.08	59.7
21	3.28	3.95	1.20	4				> 1860			42.4		59.4
22	3.82	4.33	1.13	1	5.31	1.50 $\times$ .28	1.50 $\times$ .28		1.69	6.44	63.3	3.14	
23	4.5	5.12	1.14	1	4.54	1.81 $\times$ .28	1.81 $\times$ .28		2.03	9.69	103.7	3.27	
24	3.82	4.33	1.14	2	5.0	1.50 $\times$ .28	1.50 $\times$ .28				63.3		
25	3.35	4.33	1.29	4	4.74	1.46 $\times$ .28	1.46 $\times$ .28		1.46	5.25	48.7	3.05	
26	3.82	4.33	1.14	4	5.0	1.50 $\times$ .28	1.50 $\times$ .28				63.3		
27	3.38	4.00	1.19	4	4.8	1.50 $\times$ .31	1.50 $\times$ .31	1715	1.77	4.25	45.7	3.28	85.8
28	3.38	4.00	1.19	6	4.8	1.50 $\times$ .31	1.50 $\times$ .31	1715	1.77	4.25	45.7	3.28	85.8
29	3.53	4.75	1.35	4	4.8	1.63 $\times$ .38	1.63 $\times$ .38	2050	1.86	5.38	59.2	3.32	72.3
30	4.0	4.92	1.23	4	5.0						78.7		78.3
31	3.23	4.33	1.34	4	5.0						45.2		
32	4.88	5.12	1.05	4	4.75						122		
33	3.15	4.33	1.38	4	4.0	1.18 $\times$ .28	1.18 $\times$ .28	> 1825			43.0		76.4
34	3.74	4.33	1.16	4	4.0	1.42 $\times$ .28	1.42 $\times$ .28	> 1730			60.6		81.2
35	4.33	5.12	1.18	4	4.2	1.73 $\times$ .34	1.73 $\times$ .34	> 1880			96.2		83.2
36	4.73	5.52	1.17	6	4.0	1.73 $\times$ .34	1.73 $\times$ .34	> 2290			123.6		65.0
37	4.5	6.0	1.33	1	3.5	1.75 $\times$ .44	1.75 $\times$ .44	> 1985	2.47	9.5	121.4	3.57	65.7
38	4.5	6.0	1.33	2	3.5	1.75 $\times$ .44	1.75 $\times$ .44	> 1985	2.47	9.5	121.4	3.57	65.0
39	4.5	6.0	1.33	4	3.5	1.75 $\times$ .44	1.75 $\times$ .44	> 1985	2.47	9.5	121.4	3.57	66.7
40	4.5	6.0	1.33	6	3.5	1.75 $\times$ .44	1.75 $\times$ .44	> 1985	2.47	9.5	121.4	3.57	67.0
41	6.0	8.0	1.33	4	3.5	2.50 $\times$ .56	2.50 $\times$ .63	> 1765	3.03	31.5	288	3.03	56.3
42	3.75	4.5	1.20	4	3.5	1.75 $\times$ .31	1.75 $\times$ .31	> 1725	1.96	6.0	63.2	3.25	47.8
43	4.5	5.0	1.11	4	3.78	1.75 $\times$ .44	1.75 $\times$ .44	> 2070	2.35	9.5	101.2	3.26	58.2
44e	4.5	5.0	1.11	6	3.78	1.75 $\times$ .44	1.75 $\times$ .44	> 2070	2.35	9.5	101.2	3.26	51.8
45f	4.0	7.0	1.75	4	4.26	2.25 $\times$ .63	2.25 $\times$ .63	> 1440	2.47	5.5	112	4.52	92.3
46	2.95	4.73	1.60	4	4.5	0.95 $\times$ .23	0.95 $\times$ .26	> 3800	1.48	4.6	41.2	2.99	55.3
47	3.74	4.73	1.27	4	4.5	1.26 $\times$ .25	1.26 $\times$ .28	> 2780	1.83	8.1	66.2	2.86	66.8
48	4.33	5.12	1.18	4	4.5	1.73 $\times$ .25	1.73 $\times$ .28	> 2140	2.20	10.2	96.2	3.07	65.6
49	3.54	4.73	1.34	2	3.23	1.75 $\times$ .25	1.75 $\times$ .25	1380	1.77	6.8	59.3	2.96	64.7
50	3.54	4.73	1.34	4	4.77	1.77 $\times$ .30	1.77 $\times$ .30	1350	1.83	5.94	59.3	3.16	63.5
51	3.94	5.12	1.30	4	4.42	1.97 $\times$ .296	1.97 $\times$ .296	1535	2.06	7.63	79.3	3.23	74.2
52	2.44	4.33	1.78	4	4.1	1.02 $\times$ .19	1.02 $\times$ .22	2065	1.19	3.2	25.8	2.84	53.8
53	2.56	4.33	1.69	4	4.5	1.02 $\times$ .19	1.02 $\times$ .22	2280	1.28	3.24	28.4	2.96	57.7
54	2.95	4.73	1.61	4	4.5	1.18 $\times$ .23	1.18 $\times$ .26	2470	1.50	4.43	41.2	3.05	55.2
55	4.13	4.73	1.15	4	4.84	1.75 $\times$ .31	1.75 $\times$ .31	1820	2.11	8.38	80.8	3.09	74.7
56	4.53	5.12	1.13	4	3.95	1.69 $\times$ .38	1.69 $\times$ .38		2.22	12.88	105.1	2.86	84.2
57	3.54	4.73	1.34	4	4.43	1.50 $\times$ .234	1.50 $\times$ .234	1950	1.76	6.88	59.2	2.93	66.2
58	3.25	5.0	1.54	4	5.0	1.50 $\times$ .344	1.50 $\times$ .344		1.56	7.13	52.8	2.72	86.0
59	3.5	5.0	1.43	4	5.0	1.50 $\times$ .344	1.50 $\times$ .344		1.72	7.56	61.3	2.85	84.5
60	4.0	5.0	1.25	4	5.0	1.50 $\times$ .344	1.50 $\times$ .344		1.99	9.13	80.0	2.96	86.7
61	4.25	5.0	1.18	4	5.0	1.75 $\times$ .344	1.75 $\times$ .344		2.14	10.0	90.2	3.01	83.8
62	3.15	4.73	1.50	4	4.0	1.58 $\times$ .276	1.58 $\times$ .276	1490	1.63	4.69	46.9	3.16	76.3
63	3.74	5.32	1.42	4	4.0	1.97 $\times$ .315	1.97 $\times$ .315	1820	1.96	7.0	74.3	3.26	80.5
64	3.5	4.5	1.29	4	4.76	1.625 $\times$ .22	1.625 $\times$ .25	1730	1.83	5.25	55.2	3.24	59.3
65	4.13	5.0	1.21	4	4.9	1.75 $\times$ .22	1.75 $\times$ .31	1860	2.18	7.88	85.4	3.30	59.1
66	4.75	5.0	1.05	4	4.73	2.0 $\times$ .31	2.0 $\times$ .41	1880	2.4	12.22	112.6	3.04	72.8
67	4.75	5.0	1.05	6	4.73	2.0 $\times$ .31	2.0 $\times$ .41	2350	2.4	12.22	112.6	3.04	67.2
68	3.15	4.73	1.50	4	4.0	1.42 $\times$ .28	1.42 $\times$ .28	1550	1.58	5.2	46.9	3.01	53.8
69	4.42	5.52	1.25	4	4.0	1.81 $\times$ .28	1.81 $\times$ .28	2190	2.33	9.8	107.6	3.31	52.8
70	3.56	5.12	1.44	4	5.1	1.25 $\times$ .38	1.25 $\times$ .38	2775	1.79	7.19	64.8	3.01	82.5
71	4.5	5.0	1.11	4	4.13	1.63 $\times$ .38	1.63 $\times$ .38	2220	2.27	11.1	101.2	3.02	87.3
72	5.0	5.0	1.0	4	3.96	1.63 $\times$ .38	1.63 $\times$ .38	2560	2.6	12.13	125	3.21	94.6
73	5.0	5.0	1.0	6	3.96	1.63 $\times$ .38	1.63 $\times$ .38		2.6	12.13	125	3.21	79.6
74	3.13	4.5	1.44	4	4.15	1.25 $\times$ .22	1.25 $\times$ .31	2345	1.69	3.72	44.1	3.44	54.5
75	3.56	4.75	1.33	4	4.15	1.38 $\times$ .22	1.38 $\times$ .31	2080	1.88	5.5	60.2	3.31	67.0
76	4.0	5.13	1.28	4	4.15	1.63 $\times$ .28	1.63 $\times$ .38	2575	2.1	7.63	82.2	3.28	50.0
77	3.56	5.13	1.44	6	4.15	1.38 $\times$ .22	1.38 $\times$ .31	2850	1.87	5.97	65.0	3.30	60.8
78	4.63	5.0	1.08	4	4.15	2.0 $\times$ .28	2.0 $\times$ .38	1790	2.27	13.12	107.3	2.86	66.2
79	5.0	5.13	1.03	4	4.15	2.0 $\times$ .28	2.0 $\times$ .38	1875	2.61	12.25	128.3	3.24	74.2
80	4.25	5.0	1.19	6	4.15	2.0 $\times$ .28	2.0 $\times$ .38	1655	2.27	8.68	99.2	3.38	67.7
81	4.0	3.0	0.75	4	4.22	1.38 $\times$ .31	1.38 $\times$ .31	2480	1.89	6.75	48	2.67	63.4
82	4.0	3.0	0.75	6	4.22	1.38 $\times$ .31	1.38 $\times$ .31	2110	1.89	6.75	48	2.67	74.6
83	3.13	4.75	1.52	4	3.1	1.63 $\times$ .25	1.63 $\times$ .35	1460	1.74	3.43	46.5	3.68	88.0
84	4.0	5.5	1.38	4	4.0	1.88 $\times$ .31	1.88 $\times$ .31	2490	2.39	4.87	88.0	4.25	73.6
85	4.13	5.0	1.21	1	4.17	2.0 $\times$ .22	2.0 $\times$ .25		2.21	7.31	85.3	3.42	80.2
86	3.35	4.0	1.19	4	5.0	1.44 $\times$ .30	1.44 $\times$ .33		1.66	5.25	44.8	2.92	95.7
87	4.0	4.38	1.10	2	4.0	1.63 $\times$ .32	1.63 $\times$ .35		2.05	7.25	70.1	3.11	86.7
88	4.0	4.38	1.10	4	4.0	1.63 $\times$ .32	1.63 $\times$ .35		2.01	7.81	70.1	3.0	90.2
89	3.5	4.75	1.36	4	4.09	1.38 $\times$ .31	1.38 $\times$ .31	2215			58.2		71.2
90	4.13	5.12	1.24	4	4.12	1.44 $\times$ .31	1.63 $\times$ .31				87.4		76.8
91g	5.0	5.13	1.03	4	3.85	2.13 $\times$ .38	2.13 $\times$ .38		2.29	20.5	128.2	2.5	75.2
92	4.25	5.0											

## REPORT OF HORSE POWER RATING COMMITTEE

SPEEDS.			Estimated Abs. Compression Pressure $p_v$ , lb. per sq. in.	B.H.P. BY TEST AT VARIOUS PISTON SPEEDS.															
$\sigma$ at Maximum B.H.P. Recorded.	Limiting $\sigma = 650\sqrt{\frac{d^2}{s}}$ ft. p.m.	$n$ r.p.m. corresponding to the Limiting Value of $\sigma$ .		$\sigma =$	B.H.-P.=	$\sigma =$	B.H.-P.=	$\sigma =$	B.H.-P.=	$\sigma =$	B.H.-P.=	$\sigma =$	B.H.-P.=	$\sigma =$	B.H.-P.=	$\sigma =$	B.H.-P.=		
1620	2090	2450	101	510	20	767	26.3	1025	32.0	1275	37.0	1535	41.5	1705	41.5				
1270				402	9	670	17	937	21	1205	22	1340	22						
1666				292	22	667	53	1000	76	1333	91	1500	96	1666	104				
1333	1970	2360	110	292	12	747	37	1082	45	1333	47								
1417	2140	2570	116	290	8	500	18	833	28	1166	32	1417	33						
2040	2120	2400	89	798	23.5	1063	33.5	1330	40	1595	46.5	1770	50	2040	54				
				1195	38.8														
				1025	54.3														
1082				722	16	866	18	1082	21										
1134				757	23.5	908	27.7	1134	32.5										
1140				760	27	913	30	1140	33.5										
1280				853	37	1024	43	1280	50										
1280				853	45	1024	55	1280	63										
1600				500	38	833	64	1165	80	1416	87	1600	87	1665	86				
985	2175	2610	75	840	25.5	893	26.8	960	32.0	983	32.7	1035	30						
1320				584	24	900	35	1160	37	1320	37.5								
975	1720	2760	83	437	9.6	562	12.3	762	20.1	975	22.8								
983				550	31.7	750	42.7	913	45.2	983	47.3								
955	1790	1340	62	893	190	920	198.5	933	202	955	204.5								
1200	2000	1500	62	973	97	1053	105	1120	108	1200	109								
1316				790	12	1053	16	1316	20										
			130																
			104																
			119																
			110																
			119																
1350	2130	3200	112	533	9.0	800	19.5	1067	27.8	1333	31.5	1667	27.5						
1350	2130	3200	112	533	13.5	800	28.2	1067	41.7	1333	47.2	1667	41.2						
1750	2160	2740	112	712	15.8	948	25.0	1266	33.5	1580	37.5	1900	37.5						
1475			119	575	20	900	32.5	1230	40.8	1475	44								
			119																
			111																
1025			89	810	16.0	937	17.3	1025	18.5										
1000			89	867	25.6	953	26.3	1000	27.0										
1195			94	1023	41.0	1110	43.0	1195	44.4										
1223			89	1075	60.5	1215	61.8	1223	63.6										
1220	2320	2320	75	800	7.8	1000	9.0	1100	9.4										
1200	2320	2320	75	800	15.7	1000	18	1200	18.8										
1200	2320	2320	75	800	28.3	1000	34.8	1200	38.5										
1200	2320	2320	75	800	46	1000	53	1140	57.5	1200	58.2								
1225	1970	1480	75	960	50	1040	52.5	1145	56	1225	59								
1500	2110	2820	75	600	14.5	900	19.6	1200	23.8	1500	24.0								
1250	2120	2540	83	665	25.8	1000	33.3	1250	35.0										
1500	2120	2540	83	540	30.7	665	37	833	43.1	1000	46.7								
1500	2120	2540	83	1320	46.9	1480	51.7	1560	52.8	1820	63.9								
1575	1940	2460	104	790	11	950	13	1260	16	1575	18								
1260	1860	2360	104	790	19	950	24	1260	28										
1365	2000	2350	104	855	28	1025	34	1365	40										
1350	1920	2440	68	632	8	946	12	1262	13	1420	13								
1350	2050	2600	111	632	15	946	21	1262	24	1420	25.5								
1535	2100	2460	101	855	28	1020	31	1365	35	1535	42								
1445	1845	2560	91	723	7	867	8	1155	10	1445	11								
1445	1920	2660	104	723	8.5	867	9.5	1155	11.5	1445	13.0								
1580	1980	2510	104	790	10	947	13	1260	16	1580	18								
1300	2010	2550	11	946	31.2	1025	35.2	1183	38.2	1260	39.3	1340	39.3	1420	38.4				
	1860	2180	87	853	35.0														
1400	1770	2410	102	1025	23.8	1183	26.0	1340	27.3	1420	27.6	1500	27.5	1575	26.8				
	1850	2125	119	833	18														
	1920	2220	119	833	20.5														
	1920	2310	113	833	27.5														
	1960	2350	119	833	30.0														
1500	2050	2600	89	473	9.0	867	19.2	1105	23.3	1262	25.4	1420	27.0	1577	26.8				
2018	2120	2395	89	797	24.0	1063	33.5	1328	40.5	1595	47.0	1860	53.0	2018	54.0				
1500	2110	2810	111	600	14	900	20	1200	23.6	1500	26								
1333	2140	2570	116	667	24	1000	29	1333	32										
1333	1975	2370	110	667	34	1000	46	1333	52										
1666	1975	2370	110	667	54	1000	74	1333	86	1666	90								
1260	1955	2480	89	632	10	950	13	1260	16										
1470	2150	2340	89	735	20	1105	30	1470	36										
1366	1955	2300	122	683	16	1024	26	1366	34										
1167	1960	2350	91	667	29.5	1000	43.5	1167	49.0										
1083	2090	2510	87	667	37.5	1000	57	1083	61										
	2090	2510	87	667	59	1000	71												
1500	2240	2990	93	450	8	600	11	900	15.3	1200	18.2	1500	19.0						
1250	2150	2720	93	475	13	633	17.3	950	23	1265	25.2	1425	24.0						
1710	2130	2490	93	513	16.8	683	22	1025	29	1367	31.5	1710	32.5						
1710	2140	2500	93	513	19	683	27	1025	38	1367	46	1710	47						
1333	1860	2230	93	500	23	667	29	1000	38.2	1333	44	1670	37						
1200	2110	2470	93	513	27	683	36	1025	49.3	1194	53	1367	49						
1466	2200	2400	93	550	32	733	42	1100	56	1466	64								
1180	1735	3470	95	520	16.8	675	21.0	780	23.0	870	25	1000	27	1180	28.5				
1010	1735	3470	95	438	21.2	600	28.5	675	31.5	750	33	900	39.5	1010	43				
1585	2390	3020	63	317	4	633	13.5	950	21	1266	27	1585	32.5						
2200	2760	3010	61	367	14	733	30	1100	43	1466	51.7	1833	57.5	2200	61.5				
	2220	2670	93	917	7.5	1167	9.5												
	1900	2850	119	867	22	933	23.8												
	2020	2760	89	912	14.5	1000	16.5												
	1950	2670	89	947	31.5	983	33.8												
1377			91	1030	24.7	1160	26.4	1377	28.5										
			92	768	24														
	1625	1900	85	770	34.4														
1125			95	633	13.9	842	16.3	1000	19.6	1125	20.7								
			95																
1150	1800	2740	122	460	10.9	888	23.2	1150	29.2										

*e* Marine engine. *f* Racing marine engine; connecting rod length 13 in., specially low *m*. *g* Reciprocating parts unduly heavy.

average valve setting for touring car engines, deduced from examination of forty different cases, is as follows:—

*Inlet*—Opens 12 degrees late. Closes 20 degrees late.  
*Exhaust*—Opens 45 degrees early. Closes 6 de-

**Exhaust**—Opens 45 degrees early. Closes 6 de-

grees late.

Some makers do not vary the setting for high speed work, but if the valves are small a change of setting will often cause a marked increase in the output of power. This is well shown by the following test results from a 4-cyl. 44-in. X 5 in.

engine with valves 1½ in. in diameter:—

With Inlet opening 8 in. degrees late and closing 0 degrees late, the Exhaust opening 14 degrees early and closing 0 degrees late.

the max. h.p. was 19.3 at 865 r.p.m.

With Inlet opening 13 degrees late and closing 22 degrees late, and Exhaust opening 39 degrees early and closing 10 degrees late

the max. h.p. increased to 31.2 at 1,300 r.p.m.

The value of  $\lambda$  are somewhat higher for bores of 5 in. and above, but it is probable that a smaller lift would be found sufficient in these cases on account of the diminution in surface friction with increased valve diameter. The proposed maximum rating formula equation (19) for "all out" contests may be considered inapplicable to touring car competitions where the engines are designed with reference to normal running conditions with valves usually smaller in diameter than would be required by equation (21). In such an event we may deduce a rating formula based on the limiting velocity of gas through inlet valve, assumed as 2,000 ft. per min. as follows:—

In equation (20) put  $v=2000$ ; then we have:—

the maximum practicable piston speed being now determined by valve diameter; of course, if  $\delta$  be designed in accordance with equation (21) we obtain for  $\sigma$  from (22) the same value as from equation (10).

In equation (13) substitute now for  $\sigma$  from (22) and for  $\eta p$  from (18); then, after reduction, we have:—

b.h.p. rating per cylinder =  $65^2 (1 - 1/0.85d)$  (23)  
a rating expression for touring car engines of  
simple form involving the bore and inlet valve

simple forms involving the bore and inlet valve diameter only. It resembles the Marine Motor Association rating rule; but with the addition of a factor  $(1 - 1/0.85d)$  for the  $\phi$  variation with size. The writer has not compared equation (23) with any actual test results so far, but it may prove to be worthy consideration for the purposes intended. In conclusion, it is hoped that the proposal set out in the above note, even if not found acceptable in their present form, may at least prove of assistance in enabling us to shortly reach a satisfactory rating rule for the petrol engines of automobiles.

Dugald Clerk—Well-Known English Authority—Discusses the Ratings Comprehensively

The formula recommended by the committee gives a much higher brake horsepower rating for given cylinder dimensions than the well-known R.A.C. formula, and it is accordingly necessary to explain a little more fully than is done in the report that the new rating cannot be applied indifferently to motor car engines as ordinarily found at work upon the road.

The proposed formula gives the maximum brake horsepower rating which can in the opinion of the committee be safely obtained from a petrol engine having the specified dimensions when run bolted down to a bench with every part working at its best. It by no means follows that this power is given by the same engine on the road, and it still less follows that any engine of the specified dimensions, whether old or new, could give the specified power. All that the formula means that it is possible by carefully proportioning the valves by cutting down the weights of pistons, connecting rods and other moving parts, and by a careful adjustment of the carburetor and igniting arrangements to produce the power noted by the formula. The tests of 144 engines are given in the three tables of engine data, but out of these 144 only 124 are given with sufficient particularity to calculate the value of  $\eta_p$  and the piston speed at the highest tested brake horsepower. The lowest value of  $\eta_p$  found in these tests is 47.8 lbs. per square inch and the highest is 94.6 lbs. per square inch, with the tests in which the engines gave the highest recorded brake power. The value of  $\eta_p$  thus varied from 47.8 to 94.6. Of the 124 engines 62 gave  $\eta_p$  values of 70 lbs. per square inch and less. In 36

TABLE of value diameter and lift, from Eq. (21).  $\lambda=0.28$ .

$r =$	$d =$ in.	$\delta =$ in.	$\lambda =$ in.	$r =$	$d =$ in.	$\delta =$ in.	$\lambda =$ in.
0.75	3	1.29	0.26	1.5	2.5	1.21	0.24
	4	1.82	0.36		3	1.52	0.30
	5	2.38	0.48		4	2.18	0.44
	6	2.87	0.57		5	2.77	0.55
1.0	3	1.39	0.28	1.75	2.5	1.25	0.25
	4	1.98	0.39		3	1.58	0.32
	5	2.57	0.51		4	2.22	0.44
	6	3.07	0.61				
1.25	2.5	1.16	0.23	2.0	2.5	1.28	0.26
	3	1.48	0.30		3	1.62	0.32
	4	2.08	0.42		4	2.28	0.46
	5	2.67	0.53	2.5	2.5	1.36	0.27
	6	3.26	0.65		3	1.68	0.34
				4	2.37	0.47	

engines the value lay between 80 lbs. down to 70, and 19 engines gave 90 down to 80. Half the engines thus gave results of 70 and below.

It will be observed that the proposed formula requires the value of  $\eta_p$  to vary with the cylinder diameter, so that at 2½ inches diameter  $\eta_p = 68\frac{1}{2}$  lbs. and at 5 inches  $\eta_p = 99\frac{1}{2}$  lbs. If the curve, Fig. IV, of the report, be examined it will be found that the  $\eta_p$  value given for any particular diameter of cylinder is higher on the whole than the actual values obtained by 100 out of the 124 engines. This higher value has been adopted because the committee consider that these particular pressures can be obtained in an economically working engine, although, as has been seen, comparatively few of the engines detailed in the tables of data have succeeded in obtaining these pressures.

It will be observed also that the formula is based on a piston speed varying with the stroke bore ratio from 1,200 feet per minute at  $r = 1$  and to 2,100 feet per minute at  $r = 2.5$ . These piston speeds are also considered by the committee to be safe piston speeds under certain conditions, but the tests prove that engines frequently fall below them and accordingly it is not surprising to find that engines at work on the road usually give much lower powers than would be deduced from the formula. To get an idea of the power usually exerted in actual work in a motor car upon the road by an engine of given dimensions, in my view the proposed formula should be multiplied by 0.6, that is, a little more than half the power indicated by the formula may usually be expected from the actual working engine in a motor car. This practically amounts to saying that the standard R.A.C. formula rating  $= 0.4d^3$  is correct where the stroke bore ratio is equal to 1, and this appears to me to be true, because in actual practice a piston speed of 1,000 feet per minute in ordinary work is but rarely met with, and an  $\eta_p$  value above 67 lbs. is not often found in actual work on the road, except under racing conditions. It is undoubted, however, that engines with longer stroke than bore do give higher piston speeds in ordinary work, and this is allowed for by using the proposed formula and multiplying it by .6.

When the R.A.C. formula was first proposed it gave with considerable accuracy the actual maximum brake horsepower to be expected from certain cylinder dimensions. Notwithstanding the fact that the improvements made in engines in the last five years have made it possible to obtain much higher powers under carefully arranged conditions, it yet remains true that the R.A.C. formula gives with sufficient accuracy the brake horsepower to be expected upon the road from the great mass of motor car engines now at work. If paraffin or petrol engines operating commercial vehicles be considered, the brake horsepower rating may be expected to prove to be still less. My present view is unaltered from that expressed by me in a paper read at the Royal Automobile Club

TABLES OF ENGINE DATA REFERRED TO IN

No. of Eng.	Bore in. = d.	Str. in. = s.	Str. bore Ratio $r = s/d$	No. of Cylinders = N.	Vol. Ratio of Compression.	INLET AND EXHAUST VALVES.		Mean Velocity through Inlets at Maximum H.P. = v, ft. p.m.	Value of $\delta$ given by $.29d\sqrt{\frac{d^2s}{m}}$ inches.	Mass of Reciprocating Parts in lb. = m.	$d^2s = \sqrt{\frac{d^2s}{m}}$	$\eta_p$ at Max. B.H.P. lb. per sq. in.
						Diameter and Lift of Inlets. $\delta$ in. $\times$ lin.	Diameter and Lift of Exhausts. $\delta$ in. $\times$ lin.					
95	3.75	4.5	1.20	2	5.0	1.50 $\times$ .31	1.50 $\times$ .31	1875	2.22	3.63	63.2	4.17
96	3.75	4.5	1.20	4	5.0	1.50 $\times$ .31	1.50 $\times$ .31	1875	2.18	3.8	63.2	4.03
97	3.38	4.25	1.26	4	5.0	1.38 $\times$ .31	1.38 $\times$ .31	1700	1.91	3.38	48.6	3.79
98	3.35	5.0	1.49	4	4.0	1.58 $\times$ .35	1.58 $\times$ .35		1.94	3.5	56.1	4.0
99	3.35	5.0	1.49	6	4.0	1.58 $\times$ .35	1.58 $\times$ .35		1.94	3.5	56.1	4.0
100	4.5	4.5	1.0	6	4.0	2.0 $\times$ .35	2.0 $\times$ .35	1800	2.49	6.88	91.1	3.64
101	3.15	3.54	1.12	1	3.3	1.16 $\times$ .276	1.16 $\times$ .276	2210			35.1	
102	3.15	3.54	1.12	2	3.3	1.16 $\times$ .276	1.16 $\times$ .276	2210			35.1	
103	3.15	3.54	1.12	3	3.3	1.16 $\times$ .276	1.16 $\times$ .276	2210			35.1	
104	3.15	3.54	1.12	4	3.3	1.16 $\times$ .276	1.16 $\times$ .276	2210			35.1	
105	3.15	3.54	1.12	6	3.3	1.16 $\times$ .276	1.16 $\times$ .276	2210			35.1	
106	3.35	4.33	1.29	4	3.5	1.58 $\times$ .276	1.58 $\times$ .276	1770			48.6	
107	3.94	4.33	1.10	4	3.5	1.58 $\times$ .433	1.58 $\times$ .433	2180			67.2	
108	3.94	5.12	1.30	4	3.7	1.85 $\times$ .433	1.85 $\times$ .433	2520			79.4	
109	4.33	5.12	1.18	4	4.0	1.61 $\times$ .433	1.61 $\times$ .433	2490			96.0	
110	4.73	5.12	1.08	4	4.0	1.61 $\times$ .433	1.61 $\times$ .433	2970			114.5	
111	5.0	5.12	1.02	4	4.0	2.36 $\times$ .433	2.36 $\times$ .433				128	
112	3.15	3.94	1.25	4	4.25	1.36 $\times$ .303	1.18 $\times$ .303	2340	1.56	4.63	39.1	2.91
113	3.54	4.73	1.34	4	4.25	1.50 $\times$ .315	1.50 $\times$ .315	> 1760	1.78	6.62	59.3	3.0
114	4.33	5.12	1.18	4	4.25	1.89 $\times$ .394	1.89 $\times$ .394	> 1790	2.24	9.47	96	3.19
115	5.12	5.12	1.08	4	4.25	2.36 $\times$ .473	2.36 $\times$ .473	> 1520	2.70	13.2	144.4	3.31
116	4.88	5.12	1.05	4	4.8						122	
117	3.78	5.12	1.35	4	5.0						73.2	
118	3.15	5.12	1.63	4	4.9						50.7	
119	4.0	5.0	1.25	4	4.0	2.25 $\times$ .438	2.25 $\times$ .438		2.16	6.72	80.0	3.45
120	5.0	5.25	1.05	4	5.1	2.0 $\times$ .313	2.0 $\times$ .313	> 1920	2.75	10.5	131	3.59
121	4.25	5.25	1.24	4	5.1	1.75 $\times$ .313	1.75 $\times$ .313	> 1680	2.22	9.12	94.8	3.23
122	3.15	4.5	1.43	4	5.25	1.438 $\times$ .25	1.438 $\times$ .25	> 1350	1.58	5.0	44.7	2.99
123	3.15	4.73	1.50	2	4.25	1.575 $\times$ .354	1.575 $\times$ .354	> 1260	1.74	3.53	46.9	3.65
124	3.15	4.73	1.50	4	4.25	1.575 $\times$ .354	1.575 $\times$ .354	> 1260	1.72	3.75	46.9	3.54
125	3.54	5.12	1.45	4	4.0	1.89 $\times$ .354	1.89 $\times$ .354	> 1200	1.85	6.06	64.2	3.26
126	3.54	5.12	1.45	6	4.0	1.89 $\times$ .354	1.89 $\times$ .354	> 1200	1.85	6.06	64.2	3.26
127	3.94	5.12	1.30	4	4.0	1.89 $\times$ .354	1.89 $\times$ .354	> 1300	2.04	7.72	79.4	3.21
128	3.94	5.12	1.30	6	4.0	1.89 $\times$ .354	1.89 $\times$ .354	> 1300	2.04	7.72	79.4	3.21
129	4.33	5.12	1.28	4	3.9	2.36 $\times$ .394	2.36 $\times$ .394	> 1185	2.30	9.25	103.5	3.35
130	4.92	5.92	1.20	4	3.8	2.56 $\times$ .433	2.56 $\times$ .433	> 1090	2.55	13.8	143	3.21
131	4.5	4.75	1.06	6	4.2	1.75 $\times$ .375	1.75 $\times$ .438	2150	2.37	8.75	96.2	3.32
132	4.52	4.5	1.06	4	4.0	1.375 $\times$ .375	1.375 $\times$ .375				81.2	
133	5.0	5.0	1.00	1	4.5	1.42 $\times$ .26	1.42 $\times$ .26	2500	2.20	5.3	73.5	3.72
134	3.94	4.73	1.20	1	4.2	1.42 $\times$ .26	1.42 $\times$ .26	> 3280	2.22	5.52	79.4	3.8
135	3.94	5.12	1.30	1	4.7	1.10 $\times$ .197	1.10 $\times$ .197	> 6720 ?	2.55	3.91	97.8	5.0 ?
136a	3.94	6.30	1.60	1	4.5	0.985 $\times$ .185	0.985 $\times$ .185	> 2200	1.26	3.74	26.6	2.67
137	2.60	3.94	1.52	4	4.2	1.10 $\times$ .264	1.10 $\times$ .264	2840	1.49	4.46	41.2	3.04
138	2.95	4.73	1.60	4	4.2	1.26 $\times$ .225	1.26 $\times$ .225	2560	1.76	6.95	59.3	2.93
139	3.54	4.73	1.34	4	4.2	1.65 $\times$ .225	1.65 $\times$ .225	2000	1.99	8.6	79.5	3.04
140	3.94	5.12	1.30	4	4.2	1.65 $\times$ .276	1.65 $\times$ .276	> 2810	2.53	9.87	114.5	3.41
141	4.73	5.12	1.08	4	4.2	1.26 $\times$ .315	1.26 $\times$ .315	> 2500	1.71	7.72	59.3	2.77
142	3.54	4.73	1.34	8	4.2	2.13 $\times$ .394	2.13 $\times$ .394	2740	2.90	30.4	248	2.86
143	5.92	7.08	1.20	4	4.0	1.38 $\times$ .18	1.38 $\times$ .18	> 1840	2.05	7.0	70.1	3.16
144	4.0	4.38	1.09	4	4.0							

h Reciprocating parts abnormally light and valves

TABLE I.—FIGURES FOR FOUR CYLINDERS ONLY.

No.	d =	Compression Ratio.	$\sigma =$	H. P. =	$\eta_p =$	No.	d =	Compression Ratio.	$\sigma =$	H. P. =	$\eta_p =$
52	2.44	4.1	1155	10	61.2	85	4.13	4.17	1167	38.	80.2
53	2.56	4.5	1155	11.5	63.8						
137	2.60	4.5	1050	12	71.2	61	4.25	5	833	30	83.8
46	2.95	4.5	950	13	66.1	80	4.25	4.15	1100	37.4	79.2
54	2.95	4.5	947	13	66.3	15	4.25	3.52	960	32	77.7
138	2.95	4.2	947	15.6	79.6	92	4.25	4.24	1000	39.2	91.2
						121	4.25	5.1	1050	35	77.6
74	3.13	4.13	900	15.3	73	132	4.25	—	900	27.9	72.2
83	3.13	3.1	950	21	94.8	35	4.33	4.2	1110	43	86.5
33	3.15	4	937	17.3	78.3	109	4.33	4	984	40.8	92.8
62	3.15	4	1105	23.3	89.5						
104	3.15	3.3	1082	19.7	77.2	39	4.5	3.5	1000	34.8	72.2
122	3.15	5.25	900	21.5	101.2	43	4.5	3.78	1000	46.7	96.8
						71	4.5	4.13	1000	43.5	90.2
86	3.35	5	933	23.8	95.5	56	4.53	3.95	853	35	84
106	3.35	3.5	1082	25	86.7	100	4.5	4	1050	37.5	74.2
97	3.38	5	995	19.2	71	131	4.5	4.2	990	35.3	73.9
27	3.38	4.8	1067	27.8	95.8						
						4	4.73	4.73	1082	45	78.2
2	3.5	—	937	21	76.8	12	4.73	—	1024	43	78.8
64	3.5	4.76	900	20	76	14	4.73	—	1165	53.3	86.1
89	3.5	4.09	1030	24.7	82.4	36	4.73	4	1075	40.3	70.3
29	3.53	4.8	948	25	89	110	4.73	4	984	50.6	96.5
50	3.54	4.77	946	21	74.5	141	4.73	4.2	1023	42.5	78
57	3.54	4.43	1025	23.8	77.9	66	4.75	4.73	1000	46	85.7
113	3.54	4.25	948	20	70.8						
125	3.54	4	1024	25.5	83.7	116	4.88	4.8	1025	58	99.8
139	3.54	4.2	947	23.4	83	3	4.92	—	1000	50.7	88.1
70	3.56	5.1	1024	26	84.3	130	4.92	3.8	987	44	77.5
75	3.56	4.15	950	23	80.3	72	5	3.96	1000	57	95.8
						79	5	4.15	1025	49.3	80.9
107	3.94	3.5	1082	33.7	84.3	91	5	3.85	770	34.4	75.2
108	3.94	3.7	1108	37.4	91.5	111	5	4	984	59.5	101.6
127	3.94	4	1024	30	79.3	120	5	5.1	1050	45	72
135	3.94	4.2	1023	32.8	86.8						
10	4	—	908	27.7	80						
60	4	5	833	27.5	86.7						
81	4	4.22	1000	27	70.8						
84	4	3	1100	43	102.6						
88	4	4	983	33.8	90.3						
55	4.13	4.84	1025	35.2	84.5						

TABLE II.  
Values of m., etc., from equation (1). Cast-iron pistons.

$r =$	$d,$ in inches.	$d^2s =$ $d^2r =$	$m,$ in lbs.	$d^2s/m =$	$\sqrt{d^2s/m} =$
.75	2.5	11.72	2.89	4.06	2.02
	3	20.3	3.9	5.21	2.28
	3.5	32.2	5.3	6.08	2.47
	4	48	7.2	6.67	2.58
	4.5	68.3	9.6	7.12	2.67
	5	93.8	12.6	7.44	2.73
	5.5	124.8	16.3	7.67	2.77
	6	162	20.7	7.83	2.80
1	2.5	15.63	2.94	5.32	2.31
	3	27	4	6.75	2.60
	3.5	42.9	5.4	7.94	2.82
	4	64	7.4	8.65	2.94
	4.5	91.1	9.9	9.20	3.04
	5	125	13	9.63	3.10
	5.5	166.4	16.8	9.92	3.15
	6	216	21.4	10.1	3.18
1.25	2.5	19.5	2.98	6.55	2.56
	3	33.8	4.1	8.24	2.87
	3.5	53.6	5.6	9.57	3.09
	4	80	7.6	10.52	3.24
	4.5	113.9	10.2	11.16	3.34
	5	156.3	13.4	11.67	3.42
	5.5	208	17.3	12.03	3.47
	6	270	22	12.27	3.50
1.50	2.5	23.4	3.03	7.73	2.78
	3	40.5	4.14	9.78	3.13
	3.5	64.3	5.7	11.30	3.36
	4	96	7.8	12.31	3.51
	4.5	136.7	10.4	13.14	3.62
	5	187.5	13.8	13.58	3.68
	5.5	249.6	7.8	14.03	3.74
	6	324	22.7	14.28	3.78

## REPORT OF HORSEPOWER RATING COMMITTEE

SPEEDS.			Estimated Abs. Compression Pressure $p \times 1.3 =$ const., lb. per sq. in.	B.H.P. BY TEST AT VARIOUS PISTON SPEEDS.											
$\sigma$ at Maximum B.H.P. Recorded.	Limiting $\sigma = 650 \sqrt{\frac{d^2 s}{m}}$ ft. p.m.	$n$ r.p.m. corresponding to the Limiting Value of $\sigma$ .		$\sigma =$	B.H.-P. =	$\sigma =$	B.H.-P. =	$\sigma =$	B.H.-P. =	$\sigma =$	B.H.-P. =	$\sigma =$	B.H.-P. =	$\sigma =$	B.H.-P. =
1200	2710	3610	119	600	8.1	750	10.3	900	11.6	1050	14.9	1200	15.6		
1200	2620	3490	119	600	16.3	750	20.4	900	23.0	1050	26.6	1200	29.8		
1130	2460	3470	119	567	12.2	710	15.8	850	17.7	995	19.2	1130	22.2		
	2600	3120	89												
	2600	3120	89												
1425	2365	3150	89	525	29.5	750	42.0	1050	56.2	1275	62.2	1425	63.0		
1200			70	492	2.8	787	4.1	1082	5.0	1278	5.0	1476	4.8		
1200			70	492	5.5	787	8.2	1082	10.0	1278	10.0	1476	9.6		
			70												
1300			70	492	11.1	787	16.4	1082	19.7	1278	20.2	1476	19.4		
			70												
1574			75	492	11.4	787	18.3	1082	25.0	1377	30.3	1574	30.7		
1400			75	492	18.3	787	27.4	1082	33.7	1377	36.8	1476	36.8		
1620			80	622	19.7	938	31.3	1108	37.4	1280	43.3	1450	47.3	1620	49.2
1375			89	492	21.7	787	35.2	984	40.8	1180	44.8	1377	46.3	1476	46.0
1375			89	492	29.6	787	43.8	984	50.6	1180	55.0	1377	55.8	1476	54.7
			89	984	59.5										
1315	1890	2880	96	525	9	788	13	1050	17	1315	21				
1263	1950	2470	96	632	14	948	20	1263	25						
1365	2070	2430	96	683	22	1024	30	1365	38						
1288	2150	2340	96	735	35	1103	48	1288	54						
1365			113	683	38	1025	58	1365	70						
			119												
			116												
1225	2240	2690	89	666	31.5	833	35.0	1000	43.3						
1136	2330	2660	121	525	25	700	35	875	42	1050	45	1136	46	1225	47
1125	2100	2400	121	350	13	525	19.5	700	26	875	32	1050	35	1136	36
1260	1945	2600	127	450	10	600	14	750	18	900	21.5	1050	23.5	1125	24
1260	2370	3000	96	630	5.5	788	7.5	947	8.5	1104	10	1260	11		
1260	2300	2920	96	630	12	788	15	947	16.5	1104	18	1260	20		
1365	2120	2490	89	683	17.5	853	22	1024	25.5	1194	28	1365	30		
1365	2120	2490	89	683	24	853	29	1024	34	1194	39	1365	41		
1200	2090	2450	89	683	23	853	27	1024	30	1194	32				
1200	2090	2450	89	683	32	853	38	1024	42	1194	45				
1287	2180	2370	87	736	28	920	34	1103	37.5	1287	39				
1183	2090	2120	84	788	38	987	44	1183	46						
1300	2160	2730	95	593	39	792	48	990	53	1188	55.3	1385	55.3	1583	52.5
				450	16.9	600	21.8	750	26.7	900	27.9				
				500	7.47	667	8.74	833	9.33	1000	9.56				
1300	2420	3070	104	632	5.78	947	7.56	1263	9.28	1578	8.80				
1705	2470	2900	95	682	5.6	1023	8.2	1364	9.3	1705	10.0				
2100	3250	3100	110	840	8.8	1260	13.2	1680	15.6	2100	16.0				
1314	1735	2640	104	526	6.0	788	9.6	1050	12.0	1314	12.6				
1578	1975	2500	95	632	10	947	15.6	1263	17.6	1578	18.0				
1300	1905	2420	95	632	16.5	947	23.4	1263	26.0	1578	24.0				
1400	1975	2320	95	682	20	1023	27.4	1364	31	1705	29				
1364	2220	2580	95	682	35	1023	42.5	1364	48						
1263	1800	2280	95	632	33.6	947	44.4	1263	50						
1416	1860	1575	95	1416	75.2										
	2050	2820	89	735	18	875	21.6								

extremely small, data possibly inaccurate.

TABLE II.—Continued.  
Values of  $m$ , etc., from equation (1). Cast-iron pistons.

$r =$	$d$ in inches.	$d^2 s = d^2 r$	$m$ in lbs.	$d^2 s/m =$	$\sqrt{d^2 s/m} =$
1.75	2.5	27.3	3.08	8.87	2.98
	3	47.3	4.22	11.22	3.25
	3.5	75	5.8	12.93	3.60
	4	112	8	14	3.74
	4.5	159.5	10.7	14.9	3.86
	5	218.8	14.1	15.5	3.94
2	2.5	31.25	3.13	10	3.16
	3	54	4.3	12.6	3.54
	3.5	85.8	6	14.3	3.78
	4	128	8.2	15.6	3.95
2.5	2.5	39.1	3.22	12.15	3.49
	3	67.5	4.5	15.1	3.89
	3.5	107.2	6.2	17.3	4.17

TABLE IIIA.,  
showing comparison of calculated with actual mass of reciprocating parts from the formula:  
 $m = 0.05 d^3 (1 + 0.15r) + 1.5$  in lb.  
(For pressed steel pistons.)

$d =$ inches.	$s =$ inches.	$r = s/d$	$d^2 s =$	$m$ actual.	$m$ calculated.
3.13	4.75	1.52	46.5	3.43	3.38
3.15	4.73	1.5	46.9	3.53	3.41
3.38	4.25	1.26	48.6	3.38	3.8
3.35	5	1.49	56.1	3.5	3.8
3.75	4.5	1.2	63.2	3.63	4.6
3.75	4.5	1.2	63.2	3.88	4.6
3.94	4.73	1.2	73.5	5.3	5.1
3.94	5.12	1.3	79.3	5.52	5.15
3.94	6.3	1.6	97.8	3.91?	5.28
4	5.5	1.38	88	4.87	5.12
4	7	1.75	112	5.5	5.53

on March 22, 1906. There I said: "Personally, I fear it is impossible to devise a rating rule which will enable one to accurately estimate the power of any engine from cylinder dimensions only. To obtain any such accurate rule would require uniformity of mean pressures, cylinder proportions, piston speeds and engine revolutions, which would tend in my view to impede progress rather than assist it."

The collection of data appended to the report clearly proves that even to-day petrol engine constructors have not arrived at uniformity in mean pressures, cylinder proportions, or piston speeds, and accordingly that no rule can be formed which will accurately express the power which can be given by any engine on the road from bore and stroke alone.

The mean effective pressure which corresponds to the brake horsepower developed by a petrol engine at a given piston speed has been called for some time  $\eta p$ — $\eta$  representing the mechanical efficiency of the engine, and  $p$  the mean indicated pressure upon the piston. For the purpose of the report it has been taken for the reasons given that  $\eta p$  increases with the dimensions of the cylinder. In previous discussions this increase has been considered mostly due to diminution of cooling loss with increased cylinder dimensions, and Prof. Callendar, in his paper to the Institution some years ago, has discussed the change from that point of view. It is clear, however, from the tables of engine data that the  $\eta p$  variation depends on other things as well as cooling. For example, undoubtedly mechanical efficiency changes with increased cylinder dimensions. This was clearly proved in the Institution of Civil Engineers' tests with three gas engines of 5½-inch, 9-inch and 14-inch cylinder diameter, respectively. The mechanical efficiency of those three engines was respectively .84, .85 and .86. No doubt a similar change takes place with change of cylinder diameter in petrol engines. Probably the variation of mechanical efficiency for the range covered by the formula—2½ inches to 5 inches—is greater than in the case of the three gas engines, but sufficient experiments have not been made to prove that this is so.  $\eta p$  will also vary with the total charge admitted to the cylinder, and thus with the same engine a higher value may often be obtained at a lower piston speed. This is clearly shown in the first table of engine data, where engines were tested at .9 of the maximum recorded brake horsepower. In the case of engine No. 34, for example,  $\eta p$  rises from 81.2 to 104.3 lbs. per square inch with a change of piston speed of from 1,000 to 700 feet per minute, and in engine No. 36 it rises from 65 to 100 lbs. per square inch with a change of piston speed of from 1,223 feet per minute to 715 feet per minute. Change in value of  $\eta p$  therefore obviously depends on other things as well as change of cylinder dimensions. Considerable change can also be made by varying the compression ratio, although the tables of data do not prove this.

TABLE III.,

showing comparison of calculated with actual mass of reciprocating parts from the formula:  
 $m = 0.08 d^3 (1 + 0.15r) + 1.5$  in lb.  
(For cast-iron pistons.)

$d =$ inches.	$s =$ inches.	$r = s/d$	$d^2 s =$	$m$ actual.	$m$ calculated.
2.44	4.33	1.78	25.8	3.2	2.97
2.56	4.33	1.69	28.4	3.24	3.18
2.95	4.73	1.6	41.2	4.6	4.04
3.13	4.5	1.44	44.1	3.72	4.48
3.15	3.94	1.25	39.1	4.63	4.47
3.50	4.5	1.29	55.2	5.25	5.6
3.56	4.75	1.33	60.2	5.5	5.8
3.74	4.73	1.27	66.2	8.1	6.5
4	4.38	1.1	70.1	7.25	7.5
4	5	1.25	80	6.72	7.6
4.25	5.25	1.24	94.8	9.12	8.8
4.33	5.52	1.28	103.5	9.25	9.25
4.5	5	1.11	101.2	9.5	10
4.5	6	1.33	121.4	9.5	10.25
4.63	5	1.08	107.3	13.12	10.75
4.73	5.12	1.08	114.5	9.87	11.35
4.75	5	1.05	112.6	12.22	11.4
4.92	5.92	1.2	143	13.8	12.7
5	5.25	1.05	131	10.5	13.1
5	5.13	1.03	128.2	12.25	13
5.12	5.52	1.08	144.4	13.2	13.8
5.92	7.08	1.2	248	30.4	21
7.25	7.5	1.03	394	32.5	36.7
8	8	1	512	54	48.5
12	8	.67	1152	151	153.5

Another cause may produce variations in  $\eta_p$ . If an engine be worked with considerable excess of petrol so as to produce large quantities of carbonic oxide and other unburned gases in the exhaust, then  $\eta_p$  will be increased to quite a large extent. This is proved in actual practice and may be clearly seen from Dr. Watson's experiments, but it can be inferred from the nature of petrol. Petrol is so constituted that on complete combustion a certain moderate expansion occurs, whereas in coal gas a contraction takes place. If combustion be incomplete by reason of excess of petrol a still greater expansion occurs.

It is thus evident that there are several causes leading to variation in the value  $\eta_p$ , and it must not be taken that this variation is considered to be entirely due to cooling. If the mixture in a petrol engine, however, be adjusted for maximum economy, then  $\eta_p$  will vary in a more regular manner. If an engine maker, however, desires to increase  $\eta_p$  without reference to economical working he can do so even in a small cylinder by introducing more petrol than the oxygen can burn.

The report and the tables undoubtedly prove increase of piston speed with increasing stroke bore ratio, but it should be noted that stroke bore ratio diminishes with increasing cylinder diameter, and the formula must not be taken as applicable to engines smaller than  $2\frac{1}{2}$  inches diameter or materially larger than 5 inches in diameter. The larger the diameter becomes in practice the more nearly does the stroke tend to equal the diameter.

#### Discussion

The discussion of the papers took place a week later than their reading, and there was a large number of speakers. Most of the points raised, however, were covered by Mr. Lawrence H. Pomeroy, and Mr. Lanchester, who in leading off said that the Callendar Correction was based on the assumption that, if a small engine was tested in comparison with a big one, it was impossible, owing to the more rapid cooling in the small engine, to obtain as high a mean pressure in that cylinder as in the large one. It might be asked, why not put the compression up higher so as to get a bigger output of power? The reason why they had to stop where they did was because of pre-ignition occurring, and before that a hard ignition, which was due to the fact that as the combustion proceeded through the mixture it compressed the mixture in front, so that it detonated and caused a noisy explosion. That was what limited the compression. The cure was to drop the compression slightly. That was the limit they all worked to, and he asked, was that limit the same in big as in small engines? He had considerable evidence that it was not. There was also a theoretical reason why it should not be so. As the compression curve for the small engine dropped below that for the big one, so in the small engine the loss of heat during compression fell below the curve belonging to the bigger cylinder. Consequently, a smaller combustion space would serve before the pre-ignition point was reached. Still, the engine gave what would otherwise be, except for the extra cooling, much more power than the larger engine, and these two effects corrected each other. By employing a large compression it was possible to get a higher mean pressure in a small engine than with a large one, but there was a loss by the drop in the cooling curve on the small engine. He then gave examples from Knight engine practice in illustration of this theory.

He asserted, as the result of his experience, that he was convinced it would be utterly wrong to embody this cooling correction in any rating formula. On looking at Mr. Burls's painstaking and careful piece of work one was apt to assume that all the engines were designed carefully, but he would remind them of a fact brought out in a paper by Mr. Mervyn O'Gorman, that the pistons of engines by different designers were not designed for maximum strength, but were made thicker from other considerations, and not with the greatest possible care and attention. No correction was valid unless it could be shown that the designers concerned were men of considerable ability, working independently of one another. It

was well-known how few of them were working independently.

Mr. Pomeroy said that his ambition had been to dispose of the Callendar correction, but that it seemed this matter had been covered fully by Mr. Lanchester. After some further remarks in support of Mr. Lanchester's contentions, he said that it was quite possible the formula proposed by the sub-committee for touring car engines would be used in competitions, and this formula, as Mr. Burls explained, involved a term which was equal to the cylinder diameter, minus 1.18 inches. This correction was most unfair to engines of 4 inches bore, when fitting them against engines of 3 inches bore. It meant that an engine of 4 inches bore was penalized to an extent of  $11\frac{1}{2}$  per cent. as compared with a 3-inch engine. The 84-millimeter engine was now becoming a serious factor in competitions, and it was not fair to expect an engine of 4 inches bore to develop  $11\frac{1}{2}$  per cent. more power than a 3-inch engine. As regards the series of engines made by White and Poppe and mentioned by Mr. Burls, these certainly showed an increase of mean effective pressure with size, but the small-bore engines which figured in this list were designed many years ago and calculation showed that they gave an m.e.p. which was much lower than the present day normal for small engines.

Mr. Pomeroy went on to say that Mr. Dugald Clerk, in his note, gave two or three reasons for the formula, although he did not by any means support the  $\eta_p$  correction, while giving reasons why it may be different in different engines. The first was that there is an increase in mechanical efficiency with size, on the evidence of tests made with three gas engines, but against this Mr. Pomeroy quoted some tests made by Prof. Hopkinson several years ago, in which the smaller of two particular examples was found to have the better mechanical efficiency. Mr. Pomeroy expressed the opinion that it was quite possible to get a small automobile engine as highly efficient as a large one, and went on to say that when Professor Callendar proposed the  $\eta_p$  correction it was proposed under two very rigid assumptions, the first being that the piston speed was constant and the second that the compression pressure was constant. The mixture strength was left out of the argument altogether. Therefore, he argued, the committee had chosen a formula which included a factor depending upon piston speed being constant, although the formula as a whole ignored the piston speed. Further, one reason given in support of the  $\eta_p$  correction was the result of a series of experiments by the Automobile Club of France on 96 engines. Mr. Pomeroy was disposed to give great weight to these experiments, and showed a curve plotted from their results which is reproduced herewith. From this curve it seemed that the  $\eta_p$  correction could not rationally be included in the formula. Mr. Pomeroy then said that different results could be obtained from the tables of data according to the method by which they were plotted. He suggested that Mr. Burls had had a happy accident in connection with his method of plotting, and also suggested that the  $\eta_p$  correction should not be included in the rating formula, saying that the only effect could be to produce small-bore engines with long strokes.

Continuing, Mr. Pomeroy expressed his strong disagreement with the form of the formula, which bore a resemblance to that suggested by Mr. Lanchester some years ago on the assumption that piston speed was a function of inertia stress. Working on this assumption, with reasonable piston weight and reasonable m.e.p., Mr. Pomeroy had found it would be necessary to run an engine at about 4,000 r.p.m. before the inertia stress on the neck of the connecting rod would exceed the explosion pressure on top of the piston, and he thought that this speed was outside the limits which were worthy of consideration. If this idea of piston speed were excluded it would seem quite a feasible scheme to compute horsepower on cylinder capacity only.

In conclusion, the speaker said that all engineers owed a deep debt of gratitude to Mr. Burls for collecting the data (given in the supplement published with this issue) which he considered would be of tremendous importance to

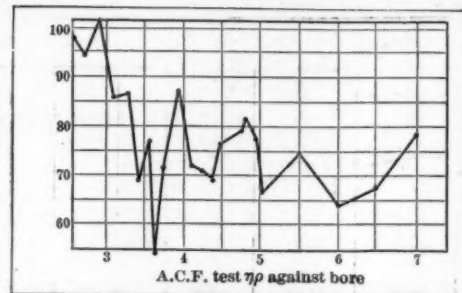


Fig. XIII—Charting the A. C. F. test  $\eta_p$  against the bore of the cylinder

the automobile engine designer in the future.

In replying, Mr. Burls said firstly, with regard to the increase of piston speed with stroke bore ratio, from considerations of gearing, comfort and so forth, the disadvantage of long strokes were more obvious in engines with large bores. Within the last two or three years an immense amount of attention had been given to engines of the 80-millimeter by 120-millimeter type, and he thought that, if an equal amount of attention was given to larger engines, they would exhibit a proportional improvement. As regards the 96 engines tested by the Automobile Club of France, Mr. Pomeroy had fallen into the error of plotting down a number of uncorrected observations. It was not fair to take the maximum horsepower, because there was a good deal of wire drawing, and tests ought to be taken before this occurred. Mr. Baillie had analyzed the French engines' results, and Prof. Callendar had examined the analysis. The committee attached great weight to the results, and in Prof. Callendar's view they found almost conclusive evidence of increase of mean pressure with size.

As regards Figs. 1, 2, and 3 in the committee's report, these were not worth consideration, because the 96 Automobile Club of France test engines were also shown at Fig. IV. of the committee's report, and though this curve fell a little below the White and Poppe curve, it was practically parallel to it. He did not think this could possibly be a mere coincidence.

#### Editorial Note

From the discussion, some of which is reported above, it appeared that many of the leading members of the Institution do not regard either the proposal of the committee or that of Mr. Burls as satisfactory. Undoubtedly, the work done is of great importance and the data gathered together by Mr. Burls is extremely valuable, but it may be questioned whether the committee have not given a little too much attention to proving the impossibility of a particular "impossible." Their aim has been to discover a reasonably simple formula which would give the power of an automobile engine in terms of bore and stroke alone, while being fairly accurate. The form of equation recommended is certainly not simple and is not extremely accurate.

The unnumbered black line in Fig. XIV is the committee rating for a four-cylinder four-inch engine with stroke varying from 2.8 to 8 inches, and immediately beneath it there is a dotted line, which is obtained from the modification of the Dendy Marshall:

$$\text{b.h.p.} = \frac{D^2 S N}{12} + 19$$

which brings the two curves together. Of course, it is not suggested that this is an actual formula, it is merely cited as showing that a simpler form might be almost as accurate as the official suggestion—quite accurate enough for ordinary purposes of comparison. It is practically the same as the contention put forward by Mr. Pomeroy that the power of normal engines varies nearly in proportion with their volumetric capacity.

(See Fig. XIV.)  
Formula.

- I. R.A.C.  $.4 D^2 S N$ .
- II. Automobile Engineer (Lanchester):  
 $1,000 D^2 N \left( \frac{2R+1}{R+2} \right)$
- III. Remington (Bourne):  $.2 D^2 S^2 \sqrt{S N}$ .
- IV. Lanchester (a):  $.5 \frac{D^2}{\sqrt{\left( \frac{D}{S} + .3 \right)}} N$ .
- V. O'Gorman (Lanchester):  $.4 D^2 S^2 N$ .
- VI. S.M.M.T.:  $.2 D N (D-1) (R+2)$ .
- Via. " :  $.33 D N (D-1) (R+2)$ .
- VII. Lanchester (b):  $.4 D^2 S^2 N$ .
- VIII. Henderson (Lanchester):  
 $.2 D N (S+D)$ .
- IX. Dendy Marshall:  $\frac{D^2 S N}{12}$ .
- X. Thornycroft:  $K D^2 S^2 N$ .
- XI. Henderson:  $.4 D^2 S^2 N$ .

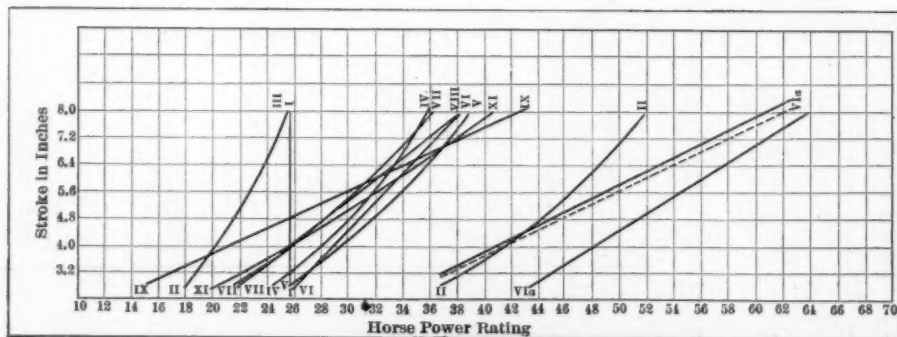


Fig. XIV—Chart of the various horsepower ratings as expressed in formulae given in the light of the stroke of the engines in inches

# It Stands to Reason—

## (Remembering That the Exception Proves the Rule)

THAT amelioration is a condition that the owners of a few junk piles would like to say "how-do-you-do" to.

THAT punishment may fit the crime that is represented in the purchase of an inferior automobile, but it is against the law to try a man twice for the same offense.

THAT responsive is not the word which can be applied to the man who pays two prices for a poor second-hand automobile when he is approached again.

THAT a distilling process would get but little "essence" out of some of the ideas that pass current in the haunts of racing.

THAT ambushing an unsuspecting purchaser is a mighty dangerous piece of business.

THAT St. Patrick could do a hard day's work banishing the snakes from the automobile grass.

THAT the dividends that some men see as coming to them if they enter the automobile business is a vapor that fills their heads at the full of the moon.

THAT sound business principles are just as much at home in the automobile business as they are in a bank.

THAT the wiles of the fellow with the gold brick in a satchel are wholly out of place if he invades the automobile business.

THAT the anatomy of the automobile situation is being patched up to the exclusion of broken bones.

THAT the seed of discontent will not sprout in a place where wisdom reclines in a chair at the head of the table.

THAT the shadow of silence cannot eclipse the situation that is always blighted by too much talk.

THAT antagonism is the fruit of a misplaced idea.

THAT Caesar was no greater than the means of transportation for which he was responsible.

THAT adversity never impedes the progress of a good idea.

THAT a man who invents a thing for which there is no demand invents nothing.

THAT the application of an idea that performs no functioning relation may be likened to a sore-thumb.

THAT inflammation is the tell-tale of a diseased condition even when it attacks the body of an automobile.

THAT too much solicitude on the part of a smiling salesman suggests the thought that it is time to go.

THAT a spurious argument accompanied by a loud voice is as empty of fact as the head of the man who presents it.

THAT apoplexy is a disorder that is not unknown to motors.

THAT boracic acid in any quantity would fail to keep poorly made tires from rotting.

THAT some of the salesmen who lie about their wares are a near approach to the human type.

THAT a purchaser who does not know what he wants is like a ship without a rudder.

THAT every automobile when it goes into service represents the beginning of a story; the ending is not always the same.

THAT a rank pretext is a poor basis for a sale.

THAT arbitrary power has a penchant for failure.

THAT abstinence is likely to desert a weak-minded salesman when he is confronted by a roll of careless money.

THAT a passion for success has the destructive qualities of a whirlwind when the owner thereof is guided by a one-sided idea.

THAT a blue Italian sky looks dull and uneventful when com-

pared with the beautiful story that accompanies some of the things that men can purchase for money.

THAT the twelve signs of the zodiac do not include the "circuit" idea that fell out of the cradle at the beginning of racing season.

THAT Heaven was never intended to be the dumping ground for a lot of bad ideas.

THAT some people seem to be lacking in a sense of discrimination.

THAT the tying of a lot of bad ideas into one bundle will not produce a good one.

THAT the raiment of a fallacy may be ever so bright and still be too scanty to clothe a fact.

THAT a mountain of a defence is insufficient to maintain the status of a thing that has no foundation in fact.

THAT the man who throws his brains away after he gets position will weigh as much as a sinker of the same size.

THAT appearance is everything until the "accountant" begins work.

THAT the principles of chemistry would apply perfectly to the activities of men.

THAT the analysis of a working idea is intended to develop the ingredients that do not make for harmony.

THAT it is in the path of wisdom to eliminate incongruities before proceeding with a plan.

THAT melancholia has an extreme fondness for the type of man who fails to fasten a poor plan onto the attention of a more vigilant neighbor.

THAT some men, were they justly paid for what they do, would owe the world enough money to sink them into bankruptcy forever.

PRODUCTION OF PIG IRON BY ELECTRICITY—On November 15, 1910, the Swedish Government commenced operations in the experimental production of pig iron by electricity on a scale sufficiently large to be considered of commercial magnitude and to approximate closely the conditions of actual commercial manufacture. Without good coal, and facing a rapid diminution in the supply of wood available for charcoal, the Swedish iron industry has felt that its salvation lies in smelting with the cheap electric energy developed from the country's abundant water power.

A contract was entered into with the directorate of the water power at Trollhattan to take 3,000 horsepower per annum for three years at \$2,680 per year, and this contract was guaranteed by the Crown. The installation is designed to produce about 20 tons of pig iron every 24 hours or, with continuous operation, 7,300 tons per annum. About 30 men are employed.

The utmost secrecy prevails regarding the operations, and no official news of the results is expected for some months. The local press, however, has stated that the operations of the experimental plant have completely fulfilled all expectations and have proved that ore can be electrically reduced with a saving of two-thirds of the coal used in the old-style blast furnaces.

There seems to be a feeling among iron manufacturers that the experiments are proving the practicability of the process, and it is reported that other furnaces are being planned. It is hoped that by the end of 1911 there will be at least four electrical blast furnaces in operation, producing at the rate of 30,000 to 35,000 tons of pig iron per year on 12,000 electric horsepower.

## S. of A. E. 1000 Strong

### Opens 1911 Summer Meeting



Henry Souther, President

*Announcing the capitulation of rule-of-thumb methods, and establishing the reign of classified knowledge, the virile members of the Society of Automobile Engineers are gathered at Dayton, Ohio, enjoying the Summer Meeting with a full program, beginning at 8:30 A.M., Thursday, June 15, with President Henry Souther in the Chair.*

CIVILIZATION'S progress may be likened to a vast military campaign in which each of the distinct elements of development occupy the positions of army corps. In the vanguard of the column that is steadily moving forward to overcome ignorance and inertia is the Transportation Corps, and the leading division is composed of the three brigades that represent the automobile industry.

On the right of the line is a picked regiment consisting of 816 specially qualified men under the command of Colonel Henry Souther, and the command is known as the Society of Automobile Engineers.

At the present moment the regiment is assembling for action at Fort Dayton, Ohio, where it is believed a spirited engagement will take place. Latest reports from the front indicate that the enemy, after a stubborn resistance, is yielding and that considerable ground will be gained as a result of the battle that is scheduled for to-day, to-morrow and Saturday. Last night a flying column, consisting of all the divisions of the Standards Squad, was thrown into the town, and after a conference as to ways and means and some discussion as to plans the force is resting on its arms at the Algonquin Hotel.

The regiment is in splendid shape, having been recruited up to its present strength by Regimental Adjutant Coker F. Clarkson, otherwise general manager of the society, and is equipped to carry on the pioneer work of its corps through another heavy campaign.

Colonel Souther's staff consists of Lieutenant-Colonel Henri G. Chatain, vice-president, and Major R. C. Carpenter, second vice-president; Adjutant Clarkson, regimental surgeon; A. H. Whiting, treasurer, and Colonels Henry Hess and H. E. Coffin, past presidents. In addition to these there are six captains detailed as aides-de-camp, namely, Hermann F. Cuntz, W. G. Wall, H. F. Donaldson, Henry May, Charles Ethan Davis and Howard Marmon. The staff is known as the Council.

The commissary department, otherwise the finance committee, is in command of Captain H. M. Swetland; the quartermaster's department, or house committee, is commanded by Captain A. J. Moulton; the ordnance department, or the regimental library committee, Lieutenant-Colonel Chatain; ammunition section, or publication committee, Captain H. F. Donaldson; signal corps, or meeting committee, Captain Swetland, and recruiting bureau, or membership committee, Captain J. G. Perrin.

The line officers, or chairmen of the various divisions of the Standards Committee, are as follows: Captain William H. Barr, in command of Company A, or the sub-division on aluminum and copper alloys; Captain David Ferguson, Company B, the ball and roller bearings division; Captain C. E. Davis, Company C, the broaches division; Captain G. G. Behn, Company D, the carbureter division; Captain James H. Foster, Company E, the frames section squad; Colonel Henry Souther, the color guard, otherwise the iron and steel division; Captain J. E. Wilson, Company F, or the lock washers division; Captain A. C. Bergmann, Company G, or the springs division, who also is acting commander of the unorganized company on the left of line, known as the miscellaneous division. Company H, or the nomenclature division, is in command of Captain P. M. Heldt, while tubes division, is in charge of Captain H. W. tion in the line is next to the colors with an effective division in the left wing, Company Company K, or the wheel dimensions and



Company I, or the seamless steel Alden. Captain Foster, whose station in the line is next to the colors with an effective division in the left wing, Company J, or the sheet metals division, fastenings for tires division, is in

# Takes Dayton by Storm

## Ringling in the New Regime

charge of Captain W. P. Kennedy, thus completing the formation.

Each of the companies has its lieutenants and non-commissioned officers and the main body of the regiment is not divided into specific divisions. The Old Guard, or main battle line, is composed of over 500 full members of the society; the second line is filled with the associate members and the squad details, sharpshooters, outposts and pickets are formed from the roster of the Juniors.

All told the society has proved to be a mobile, powerful force and its services as advance guard of the automobile division of the Transportation Corps in the grand army of civilization and progress are notable.

The march of humanity is timed to the drum-beats of the motor exhaust, and it is to this martial music that the Society of Automobile Engineers is keeping step.

The regiment is midway of its seventh year, having been organized early in 1905 for the purpose of investigation, research, discussion and publication of definite information concerning the perfection of the automobile. The first call to the colors witnessed the assembly of a mere handful of progressive, scientific men, but since then the organization has received steady additions until to-day the membership mark is approaching 1,000.

Its home armory is at 1451 Broadway, New York, but its outposts are scattered wherever there is a chance for action. Truly it represents the vanguard of civilization's march. All progress of mankind has been measured in definite terms of transportation facility. The automobile is the last word in transportation and the society is out in front of the automobile.

Just what automobile engineers have accomplished during the past decade is astounding, but a full realization of their triumph only becomes possible when specimens of current manufacture, embodying the latest proved conclusions of the engineers, are ranged alongside some of the monstrosities that served as examples of the art ten years ago. And yet the "monstrosity" was a road carriage, propelled by a gasoline engine of multiple cylinders using compression and in conjunction with a clutch device, just as those of to-day.

The difference, then, between the automobile of 1900 and that of 1912 is not to be found in the fundamental principle that covers both. The progress in manufacturing is chiefly to be noted in the improvement of the details. There is not a spot in the mechanism of the modern automobile that has not been worked over and experimented with since the days of clumsy, delicate, fragile cars of the early period. Starting with the materials from which automobiles are made, the engineers have traced out and outlined improvements in steel, aluminum, iron, copper and brass work. They have planned and designed betterments of the most detailed nature and have put them into use. The durability of the modern car is due primarily to the steel that is used in its construction and since the formation of the Society of Automobile Engineers the steel used in American cars has been radically altered from that which was used in the pioneer days. Responsibility for this measure of advancement is due solely to the work of the society through its membership.

Motor efficiency has been increased in a hundred different ways, each of which had its inception in the deliberations of the society and the working of the minds of its members. Frames have been lightened and strengthened; springs have been largely improved; gears have been changed in design and a score of mechanical processes have been made simpler. Carburetion is no longer a gamble and ignition is something better than a desperate chance. Lubrication is neither a feast nor a famine as it used to be. Externally the trim, long-lined, fore-door body of to-day bears only a faint resemblance to the rear-door tonneau effect that was so popular a few years ago, thus bearing witness to the achievements of the Automobile Engineers.

All these things mean that the balancing the details of construction to evidence of their achievements is to be

When the S. A. E. had a membership world generally. When the roll was

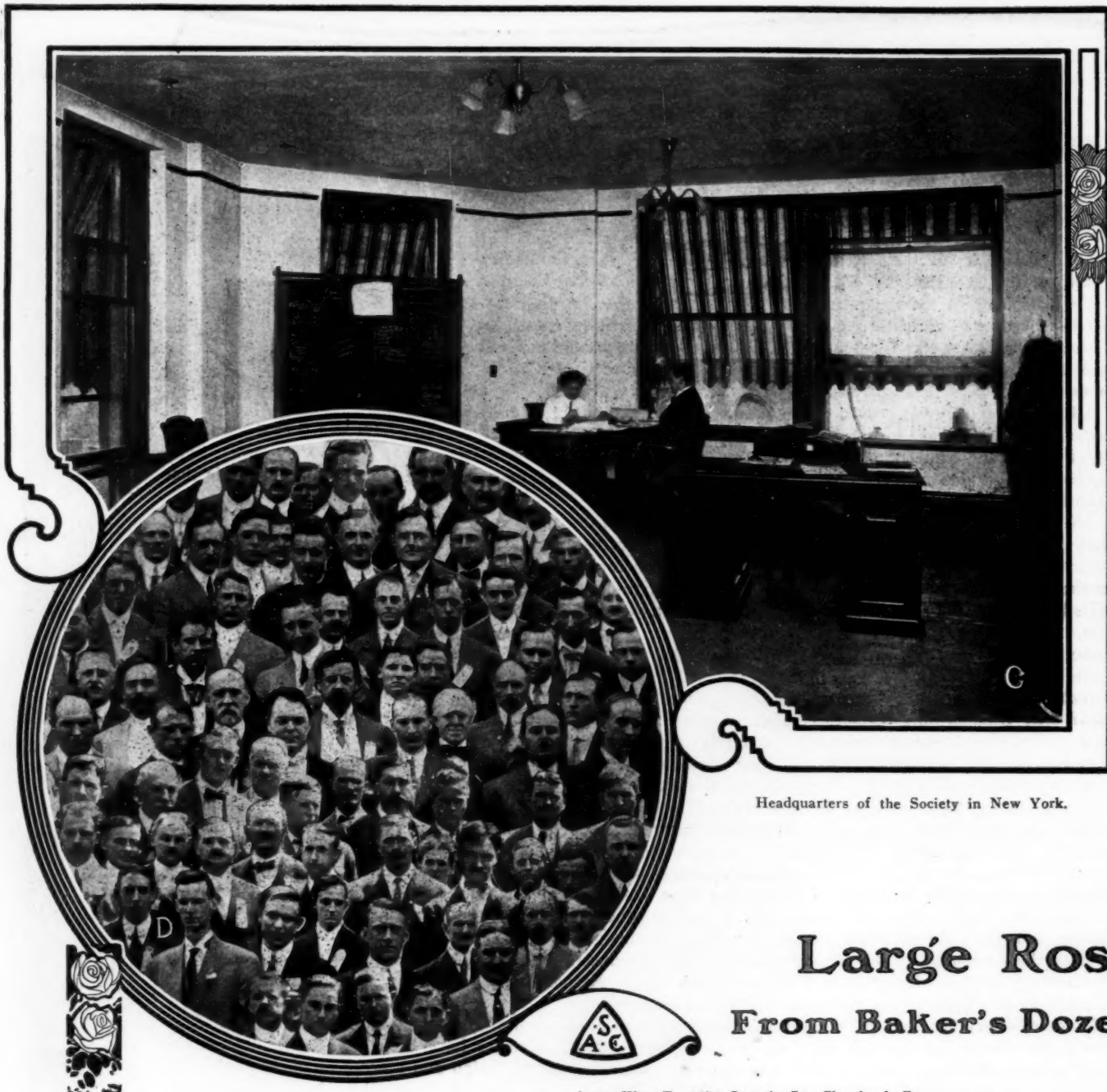


Coker F. Clarkson, General Manager.



engineers have been busy balancing and counter-produce the highest types of results. The concrete seen in the modern car.

roll of 100 it was looked upon with curiosity by the swelled to 200 the impression seemed to prevail that



Headquarters of the Society in New York.

## Large Roster

### From Baker's Dozen to

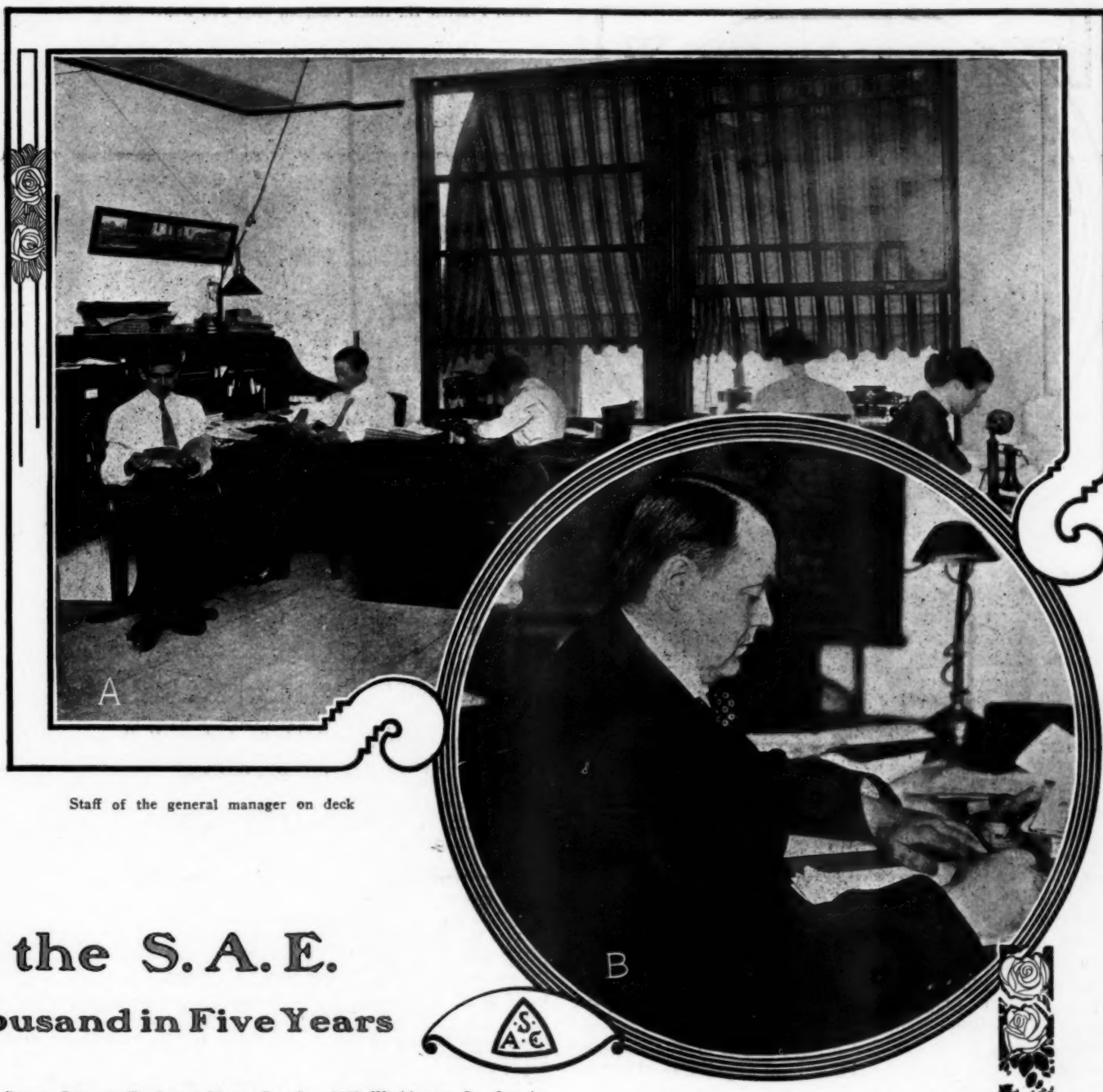
Among the "Old Guard" members

it was astonishing that so many well educated men could find anything of sufficient importance in the making of automobiles to demand their enthusiastic attention. Now that the list carries nearly 1,000 names of men who are earnestly engaged in planning detailed improvements for automobiles and in building them according to their ideas, the world sits up and takes notice.

If any engineering body whose field is limited to one direction of endeavor can command the membership of 1,000 learned scientists and active construction agencies the field must be of much importance to the race and to civilization, because never before in the history of the world has there been any such interest in such a narrow subject.

It simply means that the size of the Society of Automobile Engineers shows the gigantic measurements of the industry it represents. If the society instead of growing lustily should show a decrease in membership, it would prove that either the manufacture of automobiles had reached its climax of perfection or that the industry was entering a decline.

ABEL, WM. F., 1274 Ontario St., Cleveland, O.  
 ACKER, EMIL W., Liggett Spring & Axle Co., Cleveland, O.  
 AITKEN, W. J., 15 North 5th St., Camden, N. J.  
 AKIN, FRED. A., 716 Judson Ave., Evanston, Ill.  
 ALBORN, F. G., Locomobile Co. of America, Bridgeport, Conn.  
 ALDEN, H. W., 33 Warren Ave., E., Detroit, Mich.  
 ALEXANDER, LUDWELL B., 223 W. 46th St., New York City.  
 ALLEN, WALTER H., B. F. Goodrich Co., Akron, O.  
 ALLYNE, EDMOND E., Aluminum Castings Co., Cleveland, O.  
 ANDERSON, D. E., Clark Delivery Car Co., Grand Crossing, Chicago, Ill.  
 ANDERSON, H. B., Winton Motor Carriage Co., Cleveland, O.  
 ANDERSON, HAROLD N., Speedwell Motor Car Co., Dayton, O.  
 ANDERSON, ROSS, American Locomotive Co., Providence, R. I.  
 ANDERSON, WM. R., Anderson Forge & Machine Co., Detroit, Mich.  
 ANGER, BERNARD F., Streater Motor Car Co., Streater, Ill.  
 ANGLADA, JOSEPH A., 218 Webster Ave., Brooklyn, N. Y.  
 ANGELINO, JAS. C., 4331 Broadway, Ozone Park, N. Y.  
 APFLE, V. G., 308 N. Broadway, Dayton, O.  
 ARNOLD, EDWIN E., Metal Products Co., Detroit, Mich.  
 AULL, J. J., The Lunkenheimer Co., Cincinnati, O.  
 BACON, GEO. M., Anderson Carriage Co., Detroit, Mich.  
 BAILEY, BERTRAM, Hotel Berkeley, Minneapolis, Minn.  
 BAKER, ERLE K., Universal Kim Co., Chicago, Ill.  
 BAKER, WALTER C., American Ball Bearing Co., Cleveland, O.  
 BALDWIN, HENRY S., General Electric Co., West Lynn, Mass.  
 BALOUGH, CHARLES, Kelly Motor Truck Co., Springfield, O.  
 BANNISTER, HARRY B., Muncie Wheel Co., Muncie, Ind.  
 BARBA, WILLIAM P., Midvale Steel Co., Philadelphia, Pa.  
 BARKER, F. W., 50 Church St., New York City.  
 BARR, WM. H., Lumen Bearing Co., Buffalo, N. Y.  
 BARTLETT, E. J., Long Arm System Co., Cleveland, O.  
 BARTON, HENRY L., Metal Products Co., Detroit, Mich.  
 BASSETT, HARRY H., Weston-Mott Co., Flint, Mich.  
 BATE, JOHN W., Mitchell Lewis Motor Co., Racine, Wis.



Staff of the general manager on deck

General Manager Clarkson at work

## of the S. A. E. Thousand in Five Years

BATES, CHESTER T., Lenox Motor Car Co., 3368 Washington St., Jamaica Plains, Mass.  
 BATZELL, EUGENE P., Speedwell Motor Car Co., Dayton, O.  
 BAYLOR, A. K., 103 Park Ave., New York City.  
 BEACH, EDWARD W., Manufacturing Foundry Co., Waterbury, Conn.  
 BEE, WM. G., Edison Storage Battery Co., Orange, N. J.  
 BEECROFT, DAVID, Motor Age, Chicago, Ill.  
 BEEMER, FRANK, 1146 N. 63d St., Philadelphia, Pa.  
 BEHN, GUIDO G., Hudson Motor Car Co., Detroit, Mich.  
 BELCHER, EDWARD B., Pittsfield, Mass.  
 BENNETT, GEORGE W., Wilbys-Overland Co., Toledo, O.  
 BENTLEY, WILTON, Universal Electric Storage Battery Co., Chicago, Ill.  
 BERGMANN, ALBERT C., Fiat Automobile Co., Poughkeepsie, N. Y.  
 BERGER, FERDINAND H., Oakland Motor Car Co., Pontiac, Mich.  
 BIDDLE, WM. E., Biddle & Smart Co., Amesbury, Mass.  
 BIJUR, J., 122 60th St., New York City.  
 BILL, HARRY L., Metzger Motor Car Co., Detroit, Mich.  
 BILLINGS, F. C., Hartford, Conn.  
 BIRDSALL, E. T., Room 709 Ford Bldg., Detroit, Mich.  
 BLANCHARD, FREDERICK W., Faulkner-Blanchard Motor Car Co., Detroit, Mich.  
 BLOSS, GILBERT, Springfield Metal Body Co., Springfield, Mass.  
 BLOUGH, EARL, Aluminum Company of America, Pittsburgh, Pa.  
 BLUE, C. F., Jr., Crucible Steel Co. of America, Detroit, Mich.  
 BOOTH, CLEMENT, Standard Roller Bearing Co., Philadelphia, Pa.  
 BORNHOLT, OSCAR C., 143 Lafayette Boulevard, Detroit, Mich.  
 BOROVITZ, JOSEPH, 2422 Prospect Ave., Cleveland, O.  
 BOULIN, CAMILLE H., American Locomotive Co., Providence, R. I.  
 BOURQUIN, JAMES F., Paige Detroit Motor Car Co., Detroit, Mich.  
 BOWERSOX, GEO. G., Fort Wayne Auto-Motor Co., Fort Wayne, Ind.  
 BOWMAN, LEE H., Carnegie Steel Co., Pittsburgh, Pa.  
 BOYNTON, F. R., Providence Autogenous Welding Co., Borden & Hospital Sts., Providence, R. I.  
 BRAMAN, CHARLES E., 386 West Milwaukee Ave., Detroit, Mich.

But as each roll-call of the members discovers new names on the roster and finds the Old Guard solidly in place, it means that the industry is still growing and that perfection, like the pot of gold at the end of the rainbow, is still to be striven for.

Entirely aside from its own profits, the automobile industry has made possible even larger cities than we now have; it has spelled health to hundreds of thousands; it has proved the commercial life-preserver for many a business enterprise; it has increased with vast power the value of suburban real estate and has brought the farm into close contact with the centers of human energy. Commercially it is displacing the horse, which means a saving of millions of dollars and increased efficiency over the horse service. It also means improvement of general health by reason of the approaching elimination of old Dobbin, for the horse has been demonstrated to be a menace to health. It has made the out-of-the-way places accessible to trade, and in that measure has stimulated selling and buying and broadened the market for merchandise. But in the largest possible measure the



A. L. Riker, past president

automobile is a medium of education. It brings people into contact with one another and each time it breaks down a barrier between sections of the United States or other States in the world, in just that much it shows its most valuable purpose to humanity at large.

The automobile of to-day has been made possible by the engineers and too much credit for its present stage cannot be given to such a body as the S. A. E., and whatever may be done in the future to still further develop and improve the automobile must be credited in liberal measure to the society. To-day it stands for the motive force behind the industry and with all the well-timed precision of the machinery with which its members work it is moving forward resistlessly.

Its first chief was A. L. Riker, who served two terms as president. Mr. Riker was succeeded by Thomas J. Fay, who in turn was followed by Henry Hess. Howard E. Coffin served as executive officer of the society last year, and for 1911 Henry Souther is at its head.

Each succeeding general meeting of the society proves to be larger and more important than any of those that have preceded it and it is well within the bounds of reason to predict that the Dayton meeting will set still another new mark.

The program, which has been printed in full in *The Automobile* contains many topics of much moment to progress. Particular emphasis has been properly laid upon the broad subject of standardization and the presentation of the reports of its subdivisions will occupy the major portion of time allotted to the convention. These will be interesting as showing the exact stage at which the idea of general standardization stands to-day.

## Souther Emphasizes Standardization Work

*Henry Souther, president of the Society of Automobile Engineers, calls attention to the great work already accomplished by the organization and outlines the plans for the future. He is an ardent advocate of standardization, urges forward the work to that great end, and discusses the results of the deliberations of the various divisions of the Standards Committee.*

**T**HE Standards Committee of the Society of Automobile Engineers, when it began work about a year ago, soon found that it had a tremendous task ahead of it. Nobody ever tried to do so much standardizing before. The differ-

BREED, PRESTON H., Alden Sampson Mfg. Co., Detroit, Mich.  
 BRESEE, S. S., 311 Madison Ave., New York City.  
 BREMER, F. G., Clinton Motor Car Co., Clinton, Ont.  
 BRENNAN, PATRICK H., Brennan Motor Car Co., Syracuse, N. Y.  
 BREVFOGLE, JOHN W., 756 7th Ave., New York City.  
 BRISTOL, WM. H., The Bristol Co., Waterbury, Conn.  
 BROCKWAY, C. E., Inter-State Automobile Co., Muncie, Ind.  
 BROWN, ALEX. T., Syracuse, N. Y.  
 BROWN, H. H., 739 Boylston St., Boston, Mass.  
 BROWNLEE, D. T., The F. W. Spacke Machine Co., Indianapolis, Ind.  
 BRUSH, A. P., Buick Motor Co., Flint, Mich.  
 BUCKWALTER, TRACY V., c/o B. F. Wood, Pennsylvania Railroad, Altoona, Pa.  
 BURGESS, FRANK, Boston Gear Works, Norfolk Downs, Mass.  
 CAMERON, W. H., Willys-Overland Co., Toledo, O.  
 CARNEY, FRANK D., Pennsylvania Steel Co., Steelton, Pa.  
 CARPENTER, ROLLA C., Sibley College, Ithaca, N. Y.  
 CASE, GEORGE S., Lamson & Sessions Co., Cleveland, O.  
 CAVE, HENRY, Autogenous Welding Co., Springfield, Mass.  
 CHALMERS, HUGH, Chalmers Motor Co., Detroit, Mich.  
 CHAPIN, H. W., Browne-Lipe-Chapin Co., Syracuse, N. Y.  
 CHARAVAY, FREDERICK, Requa, Coles Co., 517 West 21st St., New York City.  
 CHASE, HERBERT, 610 W. 183rd St., New York City.  
 CHASE, J. C., 381 Fourth Ave., New York City.  
 CHASE, T. P., Room 611, Y. M. C. A., Detroit, Mich.  
 CHATAIN, HENRI G., General Electric Co., Schenectady, N. Y.  
 CHURCHWARD, ALEX., General Electric Co., 30 Church St., New York City.  
 CILLEY, RAYMOND, 1518 76th St., Brooklyn, N. Y.  
 CLAMER, G. H., Ajax Metal Co., Philadelphia, Pa.  
 CLARKE, CARL L., Driggs-Seabury Ordnance Corp., Sharon, Pa.  
 CLARK, GEORGE, Pope Mfg. Co., Westfield, Mass.  
 CLARK, JAMES H., Buick Motor Co., Flint, Mich.  
 CLARK, L. K., Bristol, R. I.  
 CLARKSON, COKER F., 1451 Broadway, New York City.  
 CLAS, ANGELO R., Falls Machine Co., Sheboygan Falls, Wis.  
 CLAYDEN, ARTHUR LUDLOW, Editor "The Automobile Engineer," 19 Great Queen St., London, W. C., England.  
 CLEMENS, C. E., 2414 Superior Ave., N. W., Cleveland, O.  
 CLENDENIN, JOHN C., 86 Ocean St., East Lynn, Mass.  
 CLOUGH, ALBERT L., 303 Kennard Bldg., Manchester, N. H.  
 COAPMAN, JOHN, E. M. F. Co., Detroit, Mich.  
 COFFIN, HOWARD E., Hudson Motor Car Co., Detroit, Mich.  
 COLBURN, HERBERT C., Colburn Automobile Co., Denver, Colo.  
 COLE, JOHN C., Fisk Rubber Co., Chicopee Falls, Mass.  
 COLER, ERNEST, Maxwell-Briscoe Motor Co., Tarrytown, N. Y.  
 COLLIER, WILLIAM H., Southern Motor Works, Nashville, Tenn.  
 COLLORD, L. W., 303 Sargeant St., Hartford, Conn.  
 CONANT, W. H., Washington Motor Vehicle Co., Washington, D. C.  
 CONOVER, EDWIN K., Watson Machine Co., Paterson, N. J.  
 CONWELL, J. S., United Motors Los Angeles Co., Los Angeles, Cal.  
 COOKE, FRANK W., 176 Sargeant St., Hartford, Conn.  
 COOKE, GEO. W., The Pierce-Arrow Motor Car Co., Buffalo, N. Y.  
 COOKINGHAM, DEWITT C., Rauch & Lang Co., Cleveland, O.  
 CORBIN, WILBUR R., W. A. Wood Auto Mfg. Co., Kingston, N. Y.  
 CORCORAN, THOMAS J., Corcoran Lamp Co., Cincinnati, O.  
 COVERT, B. V., Covert Motor Vehicle Co., Lockport, N. Y.  
 COX, CLAUDE E., 2609 Hennepin Ave., Minneapolis, Minn.  
 COX, HENRY C., Michigan Magneto Co., Detroit, Mich.  
 CRANE, HENRY M., 532 Fifth Ave., New York City.  
 CRIGHTON, PAUL L., Great Western Automobile Co., Peru, Ind.  
 CROSS, CHARLES CLARK, Chalmers Motor Co., Detroit, Mich.  
 CROSS, GEORGE S., Northway Motor & Mfg. Co., Detroit, Mich.  
 CROWLEY, JOHN A., 120 Liberty St., New York City.  
 CROXTON, HERBERT A., The Consolidated Motor Car Co., Cleveland, O.  
 CROY, HAROLD L., Inter-State Automobile Co., Muncie, Ind.  
 CUMNER, ARTHUR B., 23d and Market Sts., Philadelphia, Pa.  
 CUNEO, CHARLES, Connecticut Tel. & Elec. Co., Meriden, Conn.  
 CUNTZ, HERMANN F., 7 E. 42d St., New York City.  
 CURTIS, EDW. W., Jr., General Vehicle Co., Long Island City, N. Y.  
 DAHLQUIST, CHAS. S., International Harvester Co., Akron, O.  
 DALTON, H., Whiting Motor Car Co., Flint, Mich.  
 DABRY, JOHN, 7 E. 42d St., New York City.  
 DAVIES, E. R., 2 No. 9th St., Newark, N. J.  
 DAVIES, CHARLES ETHAN, Warner Gear Co., Muncie, Ind.  
 DAVIS, FRANK S., 386 E. Market St., Akron, O.  
 DAVIS, H. P., Westinghouse Electric & Mfg. Co., Pittsburgh, Pa.  
 DAVIS, WM. MORRIS, 556 Jefferson Ave. E., Detroit, Mich.  
 DEAN, ARTHUR M., Matheson Motor Car Co., Wilkes-Barre, Pa.  
 DEAN, PETER PAUL, Diehl Mfg. Co., 90 Prince St., New York City.  
 DECKER, FREDERICK J., Jenkins Motor Car Co., Rochester, N. Y.  
 DEEDS, EDWARD A., National Cash Register Co., Dayton, O.  
 DEJARNAY, MARCEL E., 111 Grosvenor Road, Pinlicko, London, Eng.  
 DELLING, E. H., Maytag, Mason Motor Co., Waterloo, Iowa.  
 DELREUW, ADOLPH L., Cincinnati Milling Machine Co., Cincinnati, O.  
 DEMMLER, JOHN, Clark Power Wagon Co., 509 Capitol Ave., Lansing, Mich.  
 DIETRICH, L. M., 2 Rector St., New York City.  
 DIXON, ARCHIBALD L., General Vehicle Co., Long Island City, N. Y.  
 DIXON, EDWARD, Thos. B. Jeffery Co., Kenosha, Wis.  
 DODGE, H. P., Ohio Electric Car Co., Toledo, O.  
 DONALDSON, H. F., 231 West 39th St., New York City.  
 DORRIS, GEO. P., Dorris Motor Car Co., St. Louis, Mo.  
 DOTY, A. J., Edison Storage Battery Co., Orange, N. J.  
 DOW, ALEXANDER, Dow Rim Co., 1783 Broadway, New York City.  
 DRIGGS, L. L., L. L. Driggs & Co., 90 West St., New York City.  
 DRUMMOND, R. S., 89 Marston Ave., Detroit, Mich.  
 DUCKWORTH, ARTHUR N., 14 Medford St., Boston, Mass.  
 DUESENBERG, FRED. S., 1119 West 23d St., Des Moines, Iowa.  
 DUGREY, ARTHUR, Holley Bros. Company, Detroit, Mich.  
 DUNHAM, GEO. W., Chalmers Detroit Motor Co., Detroit, Mich.  
 DUNK, ALFRED O., Autoparts Mfg. Co., Detroit, Mich.  
 DUNN, J. JAY, Shelby Steel Tube Co., Ellwood City, Pa.  
 DUNSTON, WAYNE E., Dayton Motor Car Co., Dayton, O.  
 DUPUY, HARRY W., Polack Tyre Co., Jeannette, Pa.  
 DURYEA, CHAS. E., Reading, Pa.  
 DYKE, A. L., 3974 Washington St., St. Louis, Mo.  
 EBERHARDT, F. L., Gould & Eberhardt, Newark, N. J.  
 EDISON, THOMAS ALVA, West Orange, N. J.  
 EDWARDS, FRANCIS E., Chicago School of Motoring, 1619 Michigan Ave., Chicago, Ill.  
 EDWARDS, H. J., Dayton Motor Car Co., Dayton, Ohio.  
 EHLE, A. H., Baldwin Locomotive Works, Philadelphia, Pa.  
 EHLE, EDWARD, The Crosby Co., Buffalo, N. Y.  
 EHRMAN, EDWIN H., Chicago Screw Co., Chicago, Ill.  
 ELLIS, BURTON G., 156 Park St., Medford, Mass.  
 ELLSWORTH, JOHN M., 723 Lexington Ave., New York City.  
 ENTZ, JUSTUS B., The White Company, Cleveland, O.  
 ERICKSON, C. A., Lozier Motor Co., Detroit, Mich.  
 EVELAND, S. S., 2226 Land Title Bldg., Philadelphia, Pa.

FAUROT, FAY L., C. H. Fuller Co., 510 Ford Bldg., Detroit, Mich.  
 FAY, THOS. J., *The Automobile*, 231 W. 39th St., N. Y. City.  
 FAY, WILLIAM, 1614 Chemical Bldg., St. Louis, Mo.  
 FEND, ROLAND S., Woods Motor Vehicle Co., Chicago, Ill.  
 FERGUSON, DAVID, Pierce-Arrow Motor Car Co., Buffalo, N. Y.  
 FERRIS, FRANK E., Massnick-Phipps Mfg. Co., Detroit, Mich.  
 FIELD, H. GEORGE, Sibley Quarry Co., Sibley, Mich.  
 FIRESTONE, J. F., Columbus Buggy Co., Columbus, O.  
 FLIERS, R. A., 99 Claremont Ave., New York City.  
 FOLJAMBE, E. S., 49th and Market Sts., Philadelphia, Pa.  
 FORD, BRUCE, Electric Storage Battery Co., Philadelphia, Pa.  
 FORD, HENRY, Ford Motor Co., Detroit, Mich.  
 FORREST, WM. V., Howard Motor Car Co., Galion, O.  
 FOSTER, C. H., Gabriel Horn Mfg. Co., Cleveland, O.  
 FOSTER, JAMES H., Hydraulic Pressed Steel Co., Cleveland, O.  
 FRANK, F. C., Rapid Motor Vehicle Co., Pontiac, Mich.  
 FRANQUIST, G. E., Simplex Automobile Co., 614 E. 83d St., New York City.  
 FREDERICK, WALTER A., Continental Motor Mfg. Co., Muskegon, Mich.  
 FREEMAN, LOWELL C., Federal Motor Truck Co., 554 Kirby Ave., West Detroit, Mich.  
 FRENCH, EDMUND L., Sanderson Bros. Works, Crucible Steel Co. of America, Syracuse, N. Y.  
 FRETZ, E. S., Light Mfg. & Foundry Co., Pottstown, Pa.  
 FRIEND, O. C., Mitchell-Lewis Motor Co., Racine, Wis.  
 FRITH, ARTHUR J., Armour Institute of Technology, Chicago, Ill.  
 FROELICH, CLARENCE H., Velie Motor Vehicle Co., Moline, Ill.  
 FULLER, GEORGE B., Olds Motor Works, Lansing, Mich.  
 FULLER, GEORGE F., Wyman & Gordon Co., Worcester, Mass.  
 FUNK, RICHARD W., 310 Park Ave., Weehawken, N. J.  
 FURNESS, RADCLIFFE, Midvale Steel Co., Philadelphia, Pa.  
 GAETH, PAUL, Gaeth Auto. Co., Cleveland, O.  
 GAMMETER, JOHN R. B. F. Goodrich Co., Akron, O.  
 GARCEAU, HARRY J., Warner Gear Co., Muncie, Ind.  
 GARRETSON, R. G., Bartholomew Co., Peoria, Ill.  
 GARROD, W. J., 6421 Parnell Ave., Chicago, Ill.  
 GELDART, W. EDWIN, Michican Buggy Co., Kalamazoo, Mich.  
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 GOULD, J. H., McCord Mfg. Co., Detroit, Mich.  
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Howard E. Coffin, past president

ent industries that enter into the construction of an automobile make the number of committees or divisions look very large. But there is enough work for every one of those divisions, and more. The program of the Standards Committee is not a fixed one. It cannot be until the work is older. The Society is particularly fortunate in getting the interest and assistance of good men in the industry. They have come forward and given their time and attention to correspondence and meetings in a way that is very gratifying indeed. From a more or less lukewarm enthusiast I have long since become a very warm one for the whole project of Standardization. It is needed, and there is no better proof that it is needed than the character of the correspondence and meetings that have been had.

The consumer and the producer are both interesting the committees as far as possible. The work of the committees embodies consideration of the views and necessities of both. In some cases there must be a compromise between the views of the consumer and those of the producer.

In many cases that compromise is the best thing that can be had.

**STEEL SPECIFICATION**—The specifications recommended by the Iron and Steel Division are written on a chemical rather than on a physical basis. This involves an endless discussion among engineers. The physical specification is probably not as definite as the chemical specification. It is not fair to specify both chemically and physically. It is well enough to specify impurities and their physical results, but not complete analysis as to carbon, manganese and the alloying elements, if such be used, because then the steel maker cannot always produce the goods; that is, the chemical and the physical requirements may conflict. It seems easy enough to say, "Yes, we want ten carbon and we want 40,000 elastic limit," but pretty soon along comes some engineer who says he wants ten carbon and 80,000 elastic limit. This is an exaggeration, of course, but those things happen. He may ask for twice the elongation that he ought to; perhaps that elongation is obtained at times, but to obtain it commercially is absolutely out of the question. The characteristic of a steel that may be expected may be stated, but it is impossible to lay down anything like a standard on what will happen.

It is not possible to instruct exactly as to best treatment. Sequences of operations desirable to produce certain results can be mentioned, but it cannot be said that a piece shall be quenched at exactly 1,400 degrees Fahrenheit in an oil bath. The volume of the piece cuts some figure, as likewise does the temperature of the oil, the shape of the piece and its previous history. Cov-

ering all of the various points would not constitute a group of standards, but almost a history of the state of the art, and perhaps be very confusing to those trying to use it. We could say, "Here is a specification with less than .01 per cent. phosphorus and sulphur." That is the best that can be produced commercially. It will cost money to buy it, and the improvement as regards usage in the automobile can be found in a laboratory, but hardly in any other way.

There are a certain number of steels which are used as purchased. It might be well to draw physical specifications for them. But the steelmaker will say: "You must use them as purchased; otherwise I do not guarantee results." In the case of a hardened gear the steel-maker would not accept the physical specifications and guarantee what the hardened steel would be, because that depends entirely on the treatment the gear gets.

MISCELLANEOUS MATTERS—The A. L. A. M. horsepower formula has served pretty well. At any rate it is a comparative basis, and as far as it goes is a fair measure of the relative size of two engines. Experience shows that it is not an absolute measure; that possibly a long-stroke motor may develop more power, but so far there has been no strenuous call for a change.

The question of a formula for taxation should not rest on the A. L. A. M. formula. The attitude taken now leads to a formula that will measure up real horses as well as imaginary horses, and that is going to be a difficult thing. Wide tires should be used on all wheels that are shod with steel. In Massachusetts such a law has been fairly well lived up to. In other States it is a dead letter, though it may stand on the statutes.

The question of limits of tolerance for screw threads is still in the air, and those we must have. There is a set of standard master gauges formerly belonging to the A. L. A. M. now the property of the S. A. E. But even that is incomplete as to fits—tolerance, in other words. Also relating to screws, we find that there is no standard except the United States standard for screws of more than 1 inch diameter. We are getting into that range of screws which must be covered in fine threads.

We have a pretty good precedent in the work of the Master Car Builders' Association in standardizing axle sizes for various capacities of freight cars. It is quite possible that we lack the experience at this time to duplicate that work in the freight automobile field. I look upon it as something that should be worked for, that we should correct as rapidly as possible and see what can be done in that direction. Is there any objection to standardizing the distance between spring seats and the gauge? Those seem like two very simple points. Are they as simple as they seem?

The Society of Automobile Engineers will find a fitting answer to this and many other questions. Its future is a great one. There could hardly have been more conspicuous success than that which recently attended the efforts of the Society of Automobile Engineers' Sub-committee on Wheel Dimensions for Solid Rubber Tires to untangle for the tire manufacturers, the car makers, the engineer and the user the seemingly unsolvable knots in the interchangeability of delivery wagon and truck tires.

This is but a single instance of what the Society of Automobile Engineers has done and will do in, and doubtless as a result of, its straightforward, fair-minded, far-seeing policy.

## Membership Has Shoulder to the Wheel

*Howard E. Coffin, past president of the Society of Automobile Engineers and vice-president of the Hudson Motor Car Company, presents his views of the work that is being done by the society, reflecting the good that is to be derived therefrom, and pointing out that the automobile is a virile factor in modern civilization.*

WORKING shoulder to shoulder within its membership, the Society of Automobile Engineers holds many of the brightest minds of the motor car and allied business lines. As

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an organization, the society is self-supporting, and its position is maintained without trade entanglement or commercial affiliation. The men who have worked and aided in the building up of the society have shown by their faith in the undertaking, and the persistence with which they have lent their effort that they are capable of maintaining a high ethical standard and producing practical results besides.

A little more than a decade measures the life of automobile building; it has raised from a curiosity and a joke to its position as a necessity, making it a powerful factor contributing to the marked success of present day civilization. There is no more fitting way of measuring the intensity of the activity and the progress that has been made than to simply state that the automobile business is about to take its place as the third industry in the list of the world's greatest manufacturing enterprises.

The mechanical development of the automobile is distinctly an engineering triumph, coupled with an advanced degree of scientific management, and a type of merchandizing that stands without a precedent. That the engineer represents the originating element in the industry is no more fittingly indicated than when it is pointed out that the centuries of horse-drawn wagon building paralleled by a hundred years of steamship construction during which steam railroad work advanced from its infancy to its present state of perfection offered few precedents for the guidance of the engineer along automobile lines, and he found himself in the position of having to make a practical machine out of a toy, with nothing in the previous state of the art upon which he might be permitted to lean for his guidance.

Production, methods in vogue, and the machinery equipment required in the process have been revolutionized and the best efforts of the machine tool builders have been concentrated with a view to meeting the manufacturing demand.

The motor car industry has not as yet produced its "Rankine," its "Unwin," or its "Kent." The work that is being done by the committee on standards bids fair to supply the text-book need, and a continuance of this fine effort must soon lead to a book on standards that will take its place alongside of the Rankine-Unwin-Kent text-books in the engineer's library.

The society membership is broad in its scope, and it represents a strong and progressive engineering body that is inherently capable of accomplishing many benefits to the individual engineers, the membership collectively, and the allied motor car industry.

Here's to a long and strenuous life to the Society of Automobile Engineers and success to the great work that it has before it!

## Outline of the S.A.E. Work

*Coker F. Clarkson, general manager of the Society of Automobile Engineers, takes this occasion to outline the doings of the membership, enlarging upon the activities of the Standards Committees, and relating how the society grew from a virile baby a few years ago to the proportions of an influential body of men of affairs, as it is at the present time.*

THE motto of the Society of Automobile Engineers may be said to be "Do things well and quickly." There is nothing relative about the work itself. It should be high-class, comprehensive, concise, to the point, aiding the engineers and the industry in oppressive problems. The speed with which the things can be well done is a relative matter, frequently involving laborious collection and compilation of data, interchange of ideas, and deductions which do not come easily. Ordinary technical data, such as tables, definitions and descriptions must be pertinent, accurate and authoritative. Standards recommended must be most carefully considered, in view of the future's needs and manufacturing conditions; they must be such as will be not only acquiesced in but adopted in practice; otherwise no real forward step is made.

In May, 1910, there were 289 S. A. E. members of various

grades. In January, 1911, there were 600, an increase of over 100 per cent. There are now over 900, an increase of 50 per cent. in less than six months. Applicants for membership fill out blanks giving their professional position, title or occupation, their general and technical education, including diplomas received, their record of professional experience as to both automobile and general engineering, and their specialty. They are required, except in the case of non-residents of the United States, to refer in their application to five or three members (according to grade of membership), with whom they are acquainted. These members to whom the applicants refer then communicate with the S. A. E. office, expressing their opinion as to the grade of membership to which applicants are eligible. The Society's membership committee is furnished with copies of the applications. In due course the Council of the Society, by written ballot, after being furnished with a complete summary of the above-mentioned steps, determines the membership grade eligibility of applicants, that is, the class of membership for which applicants shall be voted on by the entire S. A. E. voting membership. All engineers connected with the automobile or kindred industries are eligible for some form of S. A. E. membership. The Member grade is composed of those 26 years of age or over, who, by previous technical training or experience or by present occupation, are qualified to act as designers or constructors of complete automobile vehicles, or their component parts, or their equipment, or to exercise technical supervision of the production of materials of engineering construction; or to take responsible charge of automobile engineering work; or to impart technical instruction in automobile vehicle construction and operation.

The Associate grade consists of persons engaged in the automobile and related industries in a responsible commercial or manufacturing capacity, or who are so connected with the automobile and related industries as to be competent to co-operate with automobile engineers.

The Junior grade is made up of those under 26 years of age, who are otherwise eligible to the Member grade.

The Transactions of the Society appear in annual bound volumes, made up of the professional papers and discussion thereon presented at the two regular semi-annual national meetings of the Society, and the monthly meetings of its sections or branches which have been formed locally at important automobile centers in this country, such as New York, Philadelphia and Detroit. These Transactions are most carefully edited and fully indexed, with cross-references to the papers and remarks of each member at meetings. Alphabetical and geographical lists of members are included, together with a list of the officers and the standing, annual and professional committees; transcripts of the business meetings of the Society and financial statements.

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The professional committees above referred to are, under a provision of the Society's Constitution, appointed to investigate, consider and report upon subjects of engineering interest. Their reports by way of recommendation can be accepted by the Society. The S. A. E. Standards Committee is such a committee, and consists of about eighty members, divided into sub-committees called Divisions.

These various Divisions have been engaged in active work for over a year, and have made many reports of various kinds, partial for discussion only, and for the necessary approval of the whole Standards Committee and the Society Council, prior to recommendation to the S. A. E. membership for acceptance. Much work has been done on the following subjects: Iron and Steel, Aluminum and Copper Alloys, Solid Tire Wheel Dimensions, Broaches, Frames, Nomenclature, Lock Washers, Steel Tubing, Ball Bearings, Sheet Metal, Leaf Springs, Carburetor Connections and several important miscellaneous subjects.

## Franklin Urges 'Spanking'

Head of the H. H. Franklin Manufacturing Company suggests that the Society of Automobile Engineers establish a "Spanking Department," one of whose functions it would be to theoretically "spank" engineers who make absurd claims, after which they would be properly instructed in the dignity of the profession. He also urges standardization and the addition to the membership roll of engineers who interest themselves in the Society's activities.

I WOULD like to see the engineers of all progressive concerns become members of the Society of Automobile Engineers; not to grace the membership roll, but to work hard in the activities of the Society.

Standardization is a very important part of the work of the Society. There are also various lines of investigation which could be carried on to the benefit of all; for example, establish the facts of the practice concerning bore and stroke; say what a long motor is.

The Society could well establish a "Spanking" Department. Here the engineer of any concern who made absurd claims would be duly "spanked" and then properly instructed in the dignity of the profession.

The Engineering Department is about the most important department in any individual concern. On the engineering depends the character and thus the success of the company's output.

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 MADDEN, B. L., Kirkham Motor Mfg. Co., Bath, N. Y.  
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 MORAN, LAWRENCE R., Hudson Motor Car Co., Detroit, Mich.  
 MORGAN, JAMES B., American Automobile Mfg. Co., New Albany, Ind.  
 MYERS, ARTHUR L., McCord Mfg. Co., Detroit, Mich.  
 NEAD, JOHN H., H. H. Franklin Mfg. Co., Syracuse, N. Y.  
 OUTTEN, BURNET, 1353 Helen Ave., Detroit, Mich.  
 PARK, THOS. W., 1356 Jefferson Ave., Memphis, Tenn.  
 PRAHL, CARL A., Whiting & Dalton, Flint, Mich.  
 PRATT, WARREN CONE, United Motor Detroit Co., Detroit, Mich.  
 RADT, HUGO S., G. & A. Carburetor Co., 244 W. 49th St., New York City.  
 RICKER, CHESTER S., Sibley College, Cornell University, Ithaca, N. Y.  
 RITTER, ELMER R., The Lunkenheimer Co., Cincinnati, O.  
 ROBINSON, RICHARD T., JR., Colorado Springs, Colo.  
 SCHAEFER, C. T., 4217 No. Ninth St., St. Louis, Mo.  
 SHAPIRO, LOUIS A., Hudson Motor Car Co., Detroit, Mich.  
 SCHOENROCK, OTTO R., Lincoln Motor Car Works, Harrison & Loomis Sts.,  
 Chicago, Ill.  
 SEAMAN, FREDERICK W., S. J. Seaman, Jr., Garage, Glen Cove, N. Y.  
 STILES, WINTHROP H., 243 Columbus Ave., Boston, Mass.  
 VALENZA, VICTOR M., 229 W. 35th St., New York City.  
 VERSCHOOR, CHARLES A., Small Motor Car Co., Detroit, Mich.  
 WARNER, HENRY L., Spaulding Mfg. Co., Grinnell, Ia.  
 WARRINER, G. D., 360 Warren Ave. E., Detroit, Mich.  
 WHIPPLE, EARL B., Mercer Automobile Co., Trenton, N. J.  
 WHITE, CLARENCE L., 183 Jersey St., Buffalo, N. Y.  
 WHITE, RAYMOND E., Velie Motor Vehicle Co., Moline, Ill.  
 WINCHESTER, JOHN FROST, 773 55th St., Brooklyn, N. Y.  
 WOLF, AUSTIN M., 146 E. 81st St., New York City.  
 WOOD, CHAS. W., Cass Motor Truck Co., Port Huron, Mich.  
 WOOD, FREDERICK B., Standard Roller Bearing Co., Philadelphia, Pa.  
 WOODWARD, JAMES E., Lenox Motor Car Co., 3368 Washington St., Jamaica  
 Plain, Boston, Mass.

# Comparing Shaft-Drive Axle Types

## Distinguishing Between the Schemes of Design in Vogue

Paul Ravigneaux in "La Vie Automobile" presented a series of diagrammatic drawings and analyses of the types of live rear axles and their methods of suspension in vogue, taking the automobiles as they appeared at the Salon in 1910, and THE AUTOMOBILE presents this matter in revised and up-to-date form.

FROM the point of view of the automobile engineer, the shaft-drive type of automobile presents a problem of the first magnitude, due to the fact that the motor holds a fixed position as it

contrived at this late date. In presenting a considerable number of examples which have been collected, it may not be out of

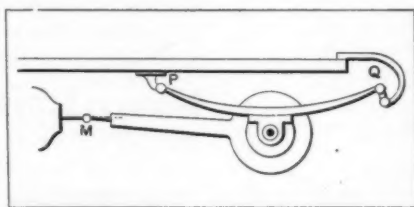


Fig. 3—B-1 type of transmission where the thrust emanates from the springs which are fixed in the front and the reaction of the couple is taken by the tubular casing or the shaft. One universal joint is used and the spring perches are permitted to articulate around the casing of the axle

the driving pinion. Supposing that one were to create this effort by hand. As the wheels resist, the pinion tries to mount the crown wheel, and it is exactly this movement that has to be avoided. It is to this effort that a reaction has to be introduced, because what is desired is not the rotation of the casing in relation to the wheels but the wheels in relation to the casing, which should remain stationary.

For this purpose one chooses the propeller shaft or its casing, the suspension springs, or a flexible joint at the end of a radius rod, or the radii rods suitably attached to support the reaction.

One must not think that it is possible to throw the thrust and the torque indiscrim-

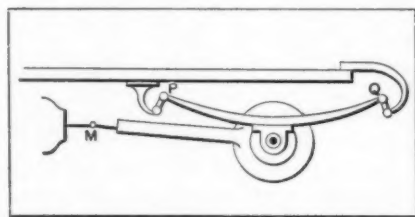


Fig. 1—Live rear axle with T casing that takes the torque and thrust efforts, the front end of the springs being shackled and one universal joint at the front end of the propeller shaft. This type is referred to as A-1

is bolted to the cross-bars of the chassis, and the live rear axle is in the relation of a variable as it is suspended on the springs and responds to the road inequalities, thus indicating that the shaft drive from the motor in its fixed relation to the driving of the traction wheels requires the use of a universal mechanism and a means for resisting "action" or, better yet, the balancing of "action" by "reaction."

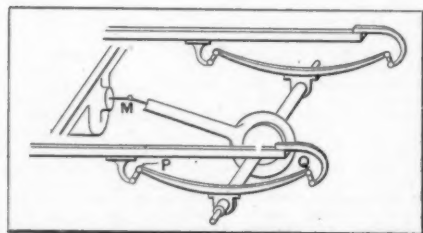


Fig. 2—Perspective view of Fig. 1 showing the amount of displacement of the whole axle and the method of spring suspension

How this task has been more or less solved by the various builders of automobiles should have more of a bearing on the future action of automobile engineers than any theoretical discussion that might be

place to say that there are faults in more or less of them, and in the automobile of the future, from the standardization point of view, it is more than likely that this important division of the designing effort will have to be given an exacting quota of attention.

An automobile must move, and in order to do so there must be something that pushes it along, and the question resolves itself into: What makes the car run along the ground? It is necessary to have an ex-

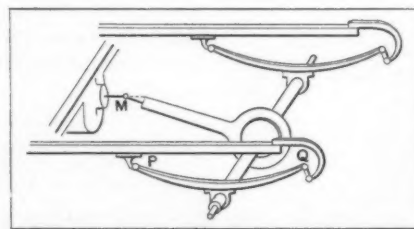


Fig. 4—Perspective view of Fig. 3 showing the displacement of the entire axle

terior force to make the car run, and this force is the reaction of the ground on the tires. It is necessary, therefore, to consider by what combination this force is transmitted to the chassis. In some cars with live axle drive it is the cardan shaft, or in many cases the tubular casing, that pushes the chassis; in other cases it is the rear springs, or again in others the radii rods do the work.

Apart from the thrust, designers have to take account of the organ that will resist the torque. The mere question of torque is not of great importance, but one must come to a clear understanding as to what takes place when the effort of the motor is transmitted to the crown wheel through

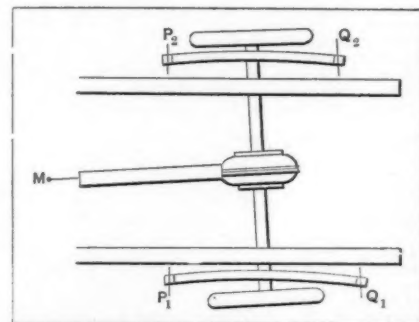


Fig. 5—Exaggerated view of the position of the axle and the springs in type B-1 transmission shown in Fig. 3 when one spring deflects more than the other. To overcome this springs with a small amount of flexure must be employed

inately on the same members. Except in exceptional cases each organ should perform its quota of the work, and there is no necessity for one to help the other except perhaps the springs; these have the right to some consideration as well, for they have a duty to perform, that of suspending the car—a duty that they must be allowed to fulfill. The subsequent remarks have no

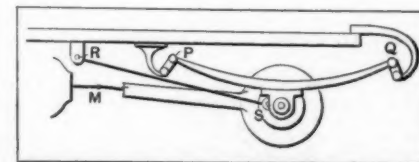


Fig. 6—In this type of transmission distance rods are used to give the necessary thrust while the torque is taken by the tubular casing of the propeller shaft. In order to obviate torsional strains in the distance rods these should be fitted with ball sockets

bearing on the type of effort that each part is called upon to support, and if one part fulfills two or three duties at the same time

—as, for example, if the springs carry the suspension, push the car as well or resist the couple of torsion or do one or the other of these things—there is nothing to be said. What should be noted, however, is that the same task should not be entrusted to dif-

ferent parts except with extreme circumstance.

In the following presentation, in order to distinguish between the schemes in vogue, a system of classification was contrived by P. Ravigneaux as follows:

THRUST.	T casing or the shaft pushes (A)	Springs push. No shackles on the front of springs (B)	Distance rods push (C).	
TORQUE.	T casing or the shaft resists the torque (1)	Spring (s) resist the torque. The perch of spring is held firm on the axle (2)	Distance rods resist the torque (3)	Torque rod resists the torque (4)

For the sake of clearness the further discussion will be in the natural order of figures. Each type of transmission will be analyzed from four points of view:

1—Considering the displacement of the entire axle.

2—Considering the vertical displacement of one wheel with the other wheel resting on the ground.

3—Considering the displacement toward the rear of one of the wheels, the other remaining in its normal position.

4—On a basis of lateral displacement of the whole axle at the expense of an equivalent movement of the chassis springs.

In order to deal with the different actions *seriatim*, the thrust organs will be designated by the letters A B C D, while the numerals 1 2 3 4 will be used to denote the resisting organs or the organs that resist the torsion or, to be more exact, the torque.

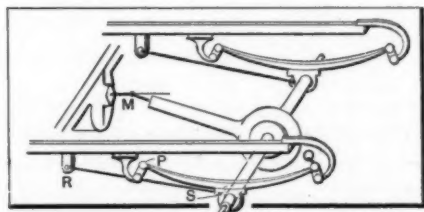


Fig. 7—Perspective view of Fig. 6, showing the displacement of the axle in example C-1. The front ends of the springs are shackled and the spring perches are not fixed solid to the axle

Supposing the shape of the casing to be a T, it is equally capable of forming the thrust as well as resisting the torque, and consequently will be found above and below the line in the foregoing table. It will be denoted by A above the line and 1 below the line. It is reasonable to suppose that if the cardan casing or the shaft

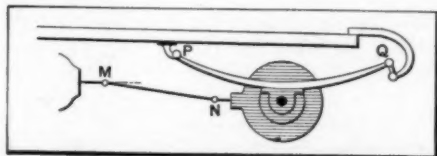


Fig. 8—The springs in this example are used as thrust and torque members as the front ends are not shackled and two universal joints are used. These permit one side of the axle to recede without affecting the alignment. This type is referred to as B-2

pushes, it is also the organ that resists the torque, therefore, there will not be any transmissions A2 or A3

The springs that are designated by B or 2 are more accommodating and they can be combined in diverse ways.

The radii rods or thrust rods have been considered as being capable of performing both rôles. Whether used for thrust or torque reaction will be considered as attached at one end to the chassis and the

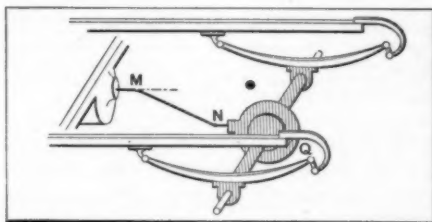


Fig. 9—Perspective view of Fig. 8, showing the displacement of the entire axle. This type has found many converts in the last few years

other to the bridge of the axle, with horizontal articulation at each end or socket.

The torque rod D or 4 must be an organ jointly related to the rotation of the bridge, consequently incapable of acting in the capacity of thrust member because the front end is fitted with an oscillating socket or slide block. If the torque rod were employed to act as thrust member it would in a way form an integral part of the bridge and would come under the classification A and not D. It would do the same work as

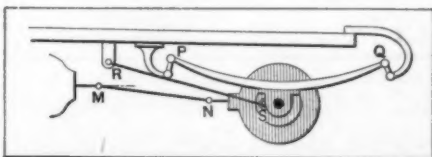


Fig. 10—Type of live rear axle in which the propeller shaft is fitted with two universal joints and the springs are shackled at the front ends as these are used to resist the torque. The thrust emanated from the distance rods, of which there are two

the T-shaped casing enclosing the propeller shaft, pushing the chassis, with the difference that the torque rod possesses usually a lateral movement in relation to the axle which the T-casing has not. The day may not be far distant when this tube also will articulate as a common torque rod, because in certain transmissions its rigidity is a detriment if not even harmful.

In the following article the difference between the central rod attached to the axle

casing at one end and to a ball socket or sliding block at the other and the side radii rods will be that the former will be designated by the appellation torque rod and the side rods as distance rods.

EXAMPLE A1—When the thrust emanates from a T-shaped case or the shaft in the absence of radius rod without compensating suspenders, and the reaction of the couple displaces the shackles T and Q, Figs. 1 and 2, coming on the universal joint M.

In this example account is taken of eight chassis of the type where the propeller shaft is assembled in a tube with a ball end, also eight chassis where the propeller shaft is assembled in a tube with a forked end.

In the further discussion distinction is made between the examples (a) with one universal joint, (b) with two universal joints and (c) without universal joints.

In this type of transmission the displacement of the entire axle will be seen by referring to Fig. 2. When one wheel is lifted from the ground the rise of the T-casing turns on its axis, due to the ball socket or the attachment of the fork. If the spring seats are not solid with the casing and have a cylindrical friction a slight torsion is produced in the springs around the points P Q. If they are attached by ball sockets this is obviated. The springs, being inherently elastic, are capable of withstanding this torsion. As the springs are shackled at the front and rear there is nothing to foresee in the case of one wheel being displaced toward the rear. It is the

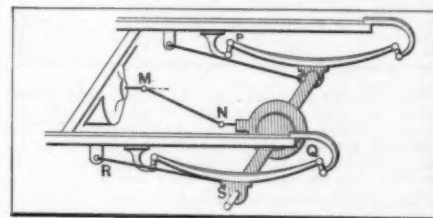


Fig. 11—Perspective view of Fig. 10 showing the displacement of the axle and the two distance rods, the spring perches are solid with the axle casing

spring that displaces fore and aft. The lateral displacement of the entire axle carries the bridge slightly crosswise because the point A is fixed. Here is a new use for the ball-socketed perches to stretch slightly from the perpendicular of the axle. If this lateral displacement is to be avoided the end of the casing must be made in the

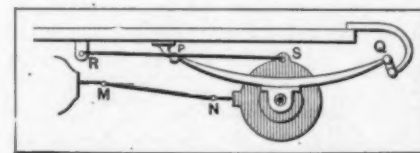


Fig. 12—Showing example B-3 transmission in which the thrust emanates through the springs which are fixed at the front ends. The single distance rod employed in the example shown takes the reaction of the couple. All kinds of displacements are possible with one rod and two universal joints, but the ends of the rod should be fitted with ball sockets

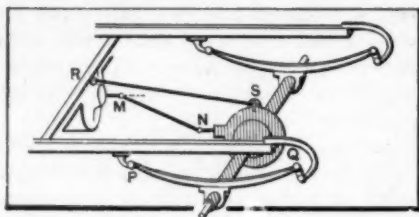


Fig. 13—Perspective view of Fig. 12 showing the attachment of the single distance rod which is placed as far back on the axle casing as possible in order to obtain the greatest amount of leverage

shape of a fork, which should be robust. Under these conditions and if the rising of the T does not cause deflections the springs are relieved of the reaction that they offer to these displacements. If softness is preferable to violence the T is mounted with a ball socket to attach it to the frame.

**EXAMPLE B-1**—When the thrust is caused by the springs, there being no torque or radii rods, and the front ends of the rear springs are fixed solid to the support without shackles, the reaction of the couple is taken by the tube or the propeller shaft. In this type there is only one cardan and the spring perches are not fixed solid to the axle casing. Eight chassis formed the basis of this type.

The displacement of the entire axle can be seen by referring to Fig. 4, and similar considerations govern the vertical displace-

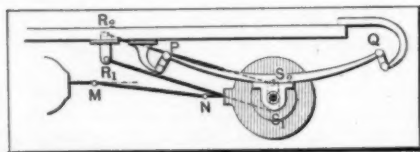


Fig. 14—Distance rods in this type, which comes under the head C-3, give the thrust and take the torque. These rods form a parallelogram on one side and the axle always remains parallel to itself. The spring perches are not fixed solid and are free to move around the axle casing

ment, as shown in A-1. The rising of the T and the shaft turn on their common axis. In this type there is slight torsion around P Q if the seats are not mounted on ball sockets. Taking into consideration the rearward displacement of a single wheel as the elongation of the spring arises from the lessening of the flexure, the bridge is caused to move obliquely (use of the ball seats). But as the tube T is not bendable the only movement permissible is a rotary one around the point M with lateral displacement of the axle.

This inconvenience can be overcome, as it is in one well-known example, by fitting the front end of the springs with rubber buffers. The rubber compresses slightly, partially compensating for the difference of the length of the springs during the time that the washers allow the lateral play that exists. In this type of transmission the axle should be able to withstand oblique movement; this necessitates the T being fitted with a ball socket as it cannot be fitted with a fork end or

with a universal, which can be in the one fitted to the shaft.

Fig. 5 is an exaggerated representation of the position of the axle and the springs in the type B-1 transmission when one spring bends more than the other. This shows the necessity of using springs with only a little bending action.

**EXAMPLE C-1**—The thrust in this case is caused by the distance rods and the torque is taken up by the T-shaped casing. In this type there is no torque rod and the springs are shackled at the front. Only one universal joint is employed and the spring

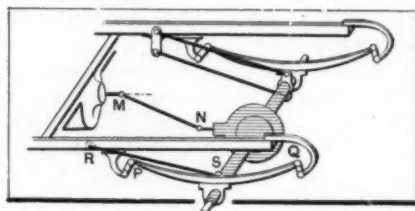


Fig. 15—Perspective view of Fig. 14, showing the method of attachment of the distance rods. The single one works for propulsion principally, doing about half this work. In this type the springs are shackled in front and the perches have free motion around the axle casing

perch is free to move on the axle casing. Eleven chassis embodying these features were seen and the type is shown in Figs. 6 and 7.

Similar conditions govern the vertical displacement of one of the rear wheels, as was shown in A-1. The rising of the T-casing and the propeller shaft turn around their common axis. There is slight torsion of the springs around P Q if the perches are not fitted with ball sockets, which shows the utility to be derived by fitting the distance rods with ball sockets so as not to have to withstand torsional efforts.

With regard to the rearward displacement of a single wheel there is nothing to foresee the spring advance and recede if

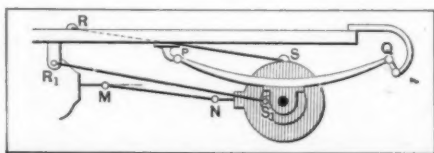


Fig. 16—This is a variable of the Fig. 14 and as will be seen there are three distance rods employed and two cardan joints. The reaction of the torque is taken by the springs as they are not shackled in the front as well as by the distance rods.

the distance rods are fitted in such a manner that the two wheels are equidistant from the point M; that is to say, that the point R projects into the point M. If this is not the case the same will be the similar to B-1, because the axle will have a tendency to shift obliquely. Slight angular displacement of the springs on their perches is to be looked for. The distance rods should be placed in such a manner that radiation of the axle around the front brackets is not more than the longitudinal play allowed by the shackles.

Considering the lateral displacement of the entire axle the distance rods prevent

the axle from moving obliquely; these work in concert with the T-casing and the springs in preventing transverse displacement of the axle to the car. It is well to allow the distance rods to have a small amount of oblique movement in relation to the plan of the chassis. Again the utility is shown of fitting these rods with ball sockets so that they work only in tension and compression.

**EXAMPLE B-2**—In this type the springs are used to form the thrust and resist the effort of the couple. There are no torque or distance rods employed, as will be seen in Figs. 8 and 9. The springs are fixed solid in the front without shackles; the spring perches are fixed solid to the axle casing, but two universal joints are employed, one at each end of the propeller shaft. Ten chassis furnished examples of this, but there must be included three others, such as the De Dion, where four transverse cardan joints are used, and the Panhard Levassor school that furnished two types. In this latter two universal joints

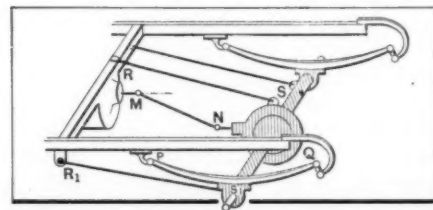


Fig. 17—The displacement of the axle as shown in Fig. 16 be seen in this illustration and the method of attaching the distance rods. The spring perches are fitted solid to the axle casing

are employed, with a T-casing with spring absorbers to resist the torque.

The total displacement of the axle can be seen in Fig. 9. The same considerations govern the vertical displacement of a single wheel, as in the preceding type, and there is a slight torsional effort around P Q which cannot be remedied as the perches cannot be fitted on ball sockets because the springs have to resist the torque. At the moment when one spring bends more than the other the half spring bends on the same side. The longitudinal spring cannot remain parallel to the outer, whence a torsion of the bridge, or, rather, as the bridge cannot give, an effort of torsion is produced around the axis of the bridge, caused by the unequal deflections of the branches of the longitudinal springs. Concerning the displacement of one wheel rearwards when the elongation of one spring is greater than the other the axle is forced farther

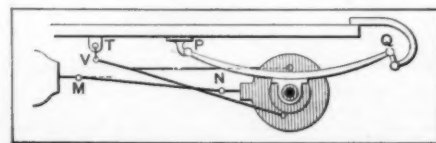


Fig. 18—In this case the thrust emanates through the springs and the reaction of the couple is taken care of by the torque rod. The spring perch is free on the axle. This type has declined in favor the T-shaped casing being cleaner in design and self-contained having less parts in the final assembly

back on that side. There being two cardans, this movement has no other effect than to cause a slight obliqueness of the springs on the perches. As designers often profit by the presence of two universal joints to give a greater degree of flexure to the springs owing to the greater obliqueness permitted to the axle in relation to the chassis (this is not advisable in type B-1, where springs with a small amount of flexure are used), and it is not possible to mount them on ball sockets, it would be good to try the mounting of the perches with an articulation around the vertical axis, which is quite practicable. One could arrange for the vertical displacement of a single wheel to be possible under these conditions as well. With regard to the lateral displacement of the whole axle the springs and their supports alone do the work.

**EXAMPLE C-2**—In this type the thrust emanates from the distance rods that can be seen in Fig. 10, and the reaction of the torque is taken up in the springs that are

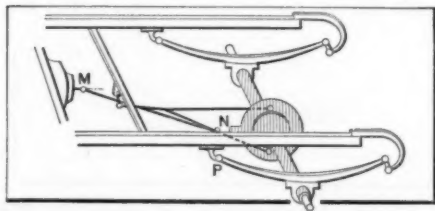


Fig. 19—Perspective view of Fig. 18, showing the cross member of the frame, to which the torque rod is attached by means of a suspension bar, and the amount of displacement of the axle when the load is imposed

fitted with front shackles. No torque rods therefore are employed in this transmission where two universal joints are employed and the spring perches are fixed solid on the casing. The vertical displacement can be seen in Fig. 11. When one wheel is displaced vertically there is produced a slight torsion of the springs around P and Q which cannot be overcome, but this has little importance. The elongation of one spring greater than the other tends to produce a torsional effort of the bridge following its axis (transverse to the car); the springs give. The phenomenon indicated in B-2 is found in all cars with fixed perches, but is a negligible quantity when there are neither transverse nor half springs at the rear. Rearward displacement of a single wheel is prevented by distance rods; it is the spring that is displaced. There is no necessity in this case to make the same

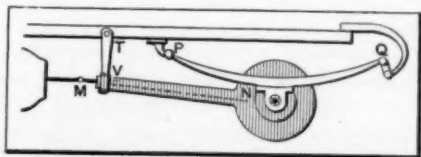


Fig. 20—This shows a variation of Fig. 18 where the casing is extended in the shape of a T belled slightly at N to house the second universal joint. The thrust emanates in this case through the springs and the torque is taken in the T casing

reservations as in C-1 and for B-1, as here we have two universal joints.

The springs and, in a small measure, the distance rods, if they are not fitted with ball sockets, resist the lateral displacement of the entire axle. It would be as well if the distance rods were not brought into play, as they would work laterally under bad conditions.

**EXAMPLE B-3**—The springs in this type are used to form the thrust members, and the reaction of the couple is taken by the

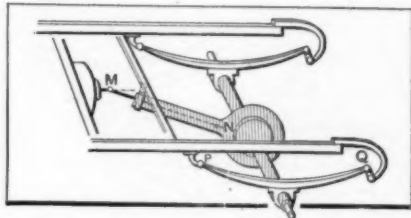


Fig. 21—Perspective view of Fig. 20 indicating the displacement of the entire axle. Only one example of this type of transmission could be found

distance rod, as there is no torque rod and the springs are attached solid to the front supports without shackles. The springs are free to rotate with their perches on the axle casing, and two universal joints are used to transmit the power to the live rear axle.

The rise and fall of the axle is accompanied by a displacement of the axle in a fore and aft sense. The center of the axle describes a certain curve. One chooses a point for attaching the distance rod to the axle as far out of the center as possible to obtain a sufficient length of leverage. From this point a curve is described not having the same center as the preceding but equal to this and with the same orientation. The center of the curve will be the point where the distance rod is attached to the chassis. By this means the displacement of the axle will be obtained without rotation, which

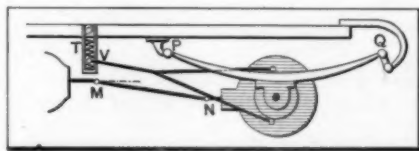


Fig. 22—This example has been styled B-2-4. The perch is fixed solid on the axle casing and the addition of the springs at the end of the torque rod permit of small displacements at the extremity of the rod. This type of rod with the springs is also found in other types of transmission and the utility of these is to soften the shocks of the clutch and brakes

will assure regularity of the transmission in the same manner where distance rods are employed to form a parallelogram.

One can be sure of the result if neither cross nor transverse springs are employed, because the more or less degree of fatigue (the change of flexure under load) of a lateral spring in relation to that of a transverse spring affects slightly the good harmony of the whole as in several other types of transmissions. The error, however, is not of great importance.

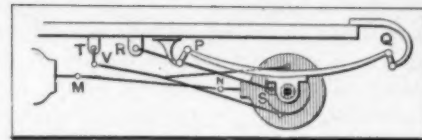


Fig. 23—This type employs separate parts to perform its specific duty. The thrust emanates from the distance rods, R S and the torque is taken care of by the torque rod V. The springs are free to move on the axle casing and are shackled at each end

When there is a vertical displacement of a single wheel a slight torsion is felt around the points P and Q of the longitudinal springs as well as in the distance rod (of reaction). To overcome this the perches and the distance rods should be fitted on ball sockets.

Considering the displacement of a single wheel toward the rear of the car as with two distance rods, this displacement, which is necessary to the freedom of the transmission, cannot be effected, as it would cause a torsional strain on the bridge; only one is fitted and that with a ball socket in the middle of the chassis, permitting all kinds of displacements. The springs re-

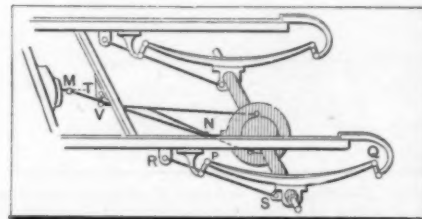


Fig. 24—This type of chassis with two universal joints in the propeller shaft, two distance rods to take care of thrust, and a separate torque rod for the reaction of the couple is found in four types of chassis. It entails very careful fitting and a quantity of wearing parts

sist the lateral displacement of the entire axle.

**EXAMPLE C-3**—This type of transmission is shown in Figs. 14 and 15, where it will be seen that the thrust is taken by the distance rods and the reaction of the couple by the distance rods also. There is no torque rod, and the springs have front shackles which allow the spring perches to be attached with free action around the casing. Two cardans are employed, as will be seen. In order that this system should be coherent it is necessary to have two distance rods that do not project one on to the other in the profile view. The displacement of the axle is effected without rotation; R1 S1 R2 S2 form a parallelogram. There are usually two distance rods on one side and one on the other. The single one does the work of propulsion. It is possible to project this, following a line parallel to R1 and S1 and R2 and S2, without intersecting these lines. Suppose this rod transmits half the effort of propulsion. Then taking the other two, one works under compression under an effort equal to the sum of the force of the couple of reaction and one-quarter of the effort of propulsion, and this is the underneath rod and the one that supports the greatest effort.

The rod located above works in tension in reaction to the couple and in compression as a push rod for one quarter of the thrust. Finally it is working all the time in tension because the reaction of the couple is always greater than the half of the thrust. During braking moments the reverse applies.

During a vertical displacement of one wheel there is a slight torsion of the springs unless they are fitted with ball socket supports. The displacement of a single wheel rearwards is not a consideration, as the springs displace and compensate for this and the displacement of the springs obviates the displacement of the entire axle. In this type the axle always remains parallel to itself, due to the parallelogram formed by the distance rods and the lines joining their points of articulation.

**EXAMPLE C-2-3**—Three chassis that were inspected showed a combination as indicated in the title of this type, in which case the reaction of the couple was taken up by the three distance rods. It will be seen in Figs. 16 and 17 from the method of

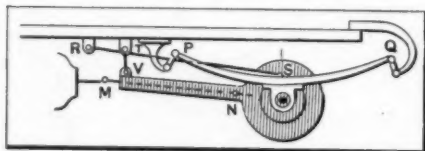


Fig. 25—This is a variable of Fig. 23 utilizing the T casing as a torque rod instead of the outside rods, as in the former case. This type of transmission will be found on the Brasier cars

attachment of the rods that the springs are not shackled at the front end. The only point to be considered here is the total displacement of the axle. If the points P and Q were permitted to move along a line P Q in guides the suspension would be absolutely free. If the right angle guides were replaced by arcs of a circle (shackles in front and rear) the spring might be slightly flanged, but this is of no moment, however. If half springs or transverse springs are added this effect is amplified more or less according to added springs taking more or less of the flexure. The springs work at the same time as the distance rods in this type of transmission to resist the torque.

**EXAMPLE B-4**—This type furnished 25 examples subdivided as follows: 18 chassis utilized the springs to transmit the thrust, using the torque rod to take the reaction of the couple; in this category the

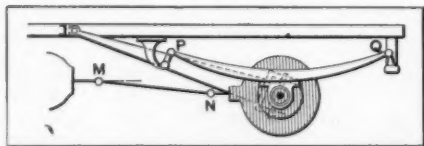


Fig. 26—This type of transmission is found in the Delaunay Belleville, and one of the two distance rods can only be seen. One of these rods is the torque member and the displacement of the entire axle is effected without interfering with the functioning of any other part

springs are not shackled in the front, two universals are utilized and the springs are permitted to articulate around the axle casing. The balance of these chassis are dealt with under the heading B-4 Variable I and B-2-4. In type B-2 Variable I, as will be seen from Figs. 20 and 21, which is comprised of only one chassis, the casing is in the form of a T which is used as the torque reaction member; two cardans serve for power transmission, one at each end of the shaft. In Variable II of this

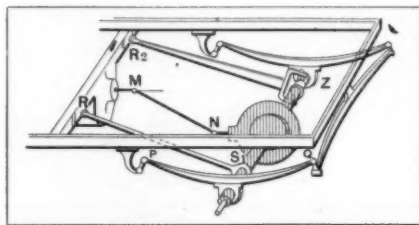


Fig. 27—Perspective view of Fig. 26, showing how the two rods that serve as torque and distance rods are attached in the front by ball sockets and at the rear at S by a ball socket and at Z by a yoke and cap

class there are 6 chassis, Fig. 22 showing how a spring absorber resists the torque. The spring perches are fixed in this case. The displacement of the axle in type B-4 can be seen in Fig. 19 and when a vertical displacement of one wheel takes place there is a slight torsion of the springs around the points P Q unless, as has been shown in previous cases, they are mounted on ball sockets. When one wheel is displaced towards the rear of the car in order to prevent the springs from deflecting on the perches these latter can be fitted with ball sockets. It is also useful to mount the torque rod in a similar manner so that it can oscillate around the vertical axis of the casing or that it be sufficiently elastic to deflect or that the forward extremity be capable of being displaced laterally (these three solutions are found in

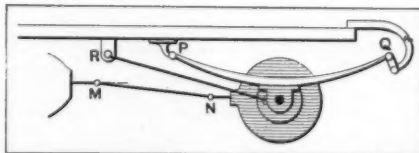


Fig. 28—In this type of transmission, unless there is sufficient play in the eyes of the spring P, the thrust taken by the rods R will inconvenience the proper working of the springs. Provided both organs are not working in propulsion at the same time this type offers good suspension

practice). When there is a tendency for the axle to be displaced the springs and their supports alone do the work.

In B-4, Variable I, the casing made out of the tube forms the torque rod which covers the second universal joint at N in Figs. 20 and 21. The one point to be taken into consideration in this type is that in order to overcome the tendency of the displacement of one of the wheels rearwards springs with a small amount of flexure should be employed. Looking at

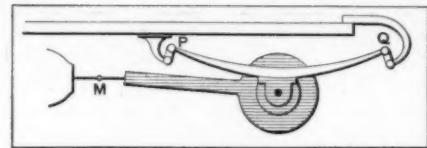


Fig. 29—In this type of transmission as shown the springs are shackled in the front and when the spring bends the front half will exfoliate

example B-2-4, as depicted in Fig. 22, and considering the question of the lateral displacement of the whole axle, this is similar to B-4 as above indicated. In the displacement of the whole axle, in order that the suspension should be as free as possible, it is necessary that the center of the ball socket should not coincide with the mean center of the curve described by the axle in relation to the chassis, but this consideration is not of the greatest amount of importance as the combination is good or bad according to whether the point V, united solid to the axle, which has a more or less vertical displacement coupled with a slight rotary motion, takes a horizontal displacement when the suspender is removed. The same considerations govern the displacement of the single wheel towards the rear

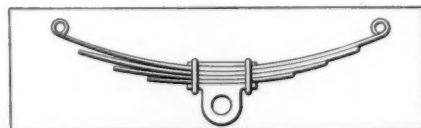


Fig. 30—This shows only in an exaggerated form what takes place in the type of transmission depicted in C-1-2. This applies in a degree in transmissions where the perches are fixed solid

as in B-2, but it is advisable to make the torque rod so that it has lateral flexure or oscillation.

Constructors of this type fix the perches solid on the axle and add springs that permit slight displacements at the extremity of the torque rod. As a matter of fact these springs often will be met in first-class transmissions of the B-4 type, but they are not necessary, although they soften the shocks that are imparted by the clutch and brakes.

**EXAMPLE C-4**—In this type of transmission the springs are shackled at the front end and have a free motion on the axle casing, the perches being fitted in such a manner as to allow this. Two universal joints are employed and the thrust is taken by the distance rods fitted at each side of the chassis. The torque is taken care of by the central torque rod. Operating along the same line, but with some alterations, as may be seen by referring to Figs. 25, 26 and 27, are the variations of this example.

C-4 example will be seen by referring to Figs. 23 and 24, showing the side and plan views of the rear end of the chassis. The displacement of the entire axle can be seen in Fig. 24. When one wheel is displaced vertically a slight torsion of the springs takes place around the points P Q, unless the perches have ball socket attachments.

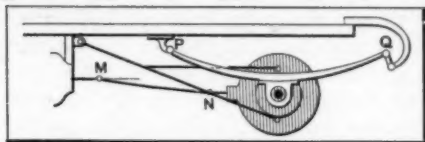


Fig. 31—In this type of transmission the springs act as thrust members and the torque rod acts in a similar capacity

Torsion of the distance rods has to be guarded against or they should be fitted with ball sockets. When one wheel tends to displace towards the rear the torque rod has little need to oscillate or bend if the point R is projected in the vicinity of M, because it is the springs that are displaced from the front to the rear or inversely. But it is possible that the oscillation or bending laterally would have regard to 4. The springs resist the lateral displacement of the entire axle.

Fig. 25 shows the example of the Variable I of C-4, in which case the T-shaped casing enclosing the propeller shaft acts

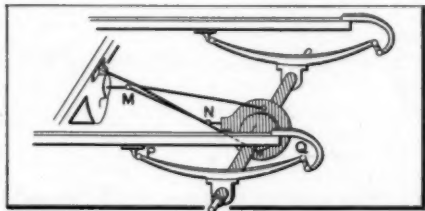


Fig. 32—Method of attachment of the springs and torque rod of Fig. 31. It is possible in this type that when the spring deflects, bending strains will be imposed on the chassis members

as torque rod. The vertical displacement of a single wheel while the other remains stationary, the displacement of a wheel rearwards, as well as the lateral displacement of the entire axle, are similar to the example C-4. The main point to be considered in this type is that to overcome the displacement of the entire axle the point R should be placed as near as possible in projection to the point M, so that the shaft has not a large amount of displacement in relation to the casing.

Variable II of C-4 is used on Delaunay Belleville cars and might as well come under the C-1 category. The displacement of the entire axle is effected in such a manner that the other organs are not interfered with. In vertical displacements of a single wheel the springs are not subjected to any torsional efforts. The distance rods do not remain parallel and a relative displacement is permissible, due to the presence of ball sockets at R-2 and R-1. The rearward displacement of a single wheel is limited by the distance rods. The springs move back and forth and the axle proper moves slightly obliquely. This is provided for as the socket S and the cap Z give the necessary play.

The longitudinal springs resist the effort of the lateral displacement of the entire axle and give sufficiently without binding any other organ.

Among the following examples will be

found six different types of chassis that the author terms dissentants. This category has been reduced since the last census was taken, in the year 1908, from 11 cars out of a total of 79 to 6 out of 89 in 1910. In examining the example B-C-2 as shown in Fig. 28 for displacement of the entire axle it will be seen that the springs and the distance rods combine in the effort of thrust. No matter how carefully the points R and S are chosen (and it must be remembered that these vary if the rear springs do not deflect according to the

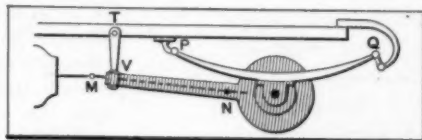


Fig. 33—This type is very similar to Fig. 31 only that the constructor extends the casing into a T form and with the aid of a drop arm utilizes it as a torque rod. Longitudinal efforts are not produced, but the springs exfoliate.

same law as the longitudinal ones) all that can be determined is that the play in the eye P of the spring and in the articulations R and S will be sufficient, so that the distance rod does not interfere with the springs. It is possible to obtain by this means an excellent suspension, provided that one of the two organs, distance rod or springs, does not come into action and support the effort of thrust or do not do the work at the same time. If there is sufficient play in the eye of the spring P this is like the example C-2.

COMBINATION A-I-2—The main consideration here is the displacement of the entire axle. When the spring deflects it should exfoliate on the one side and work in the opposite manner on the other side, because it seems difficult to subordinate the study of the car to the choice of the

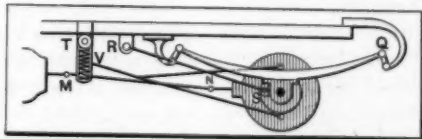


Fig. 34—There is a difficulty in this case to find a point V suitable for the combination, but this can be approximated by fixing R and S properly. The perches are free and the thrust is taken care of by the distance rods

proper position of the point M if such exists. Besides in the same case the non-proportional flattening of the back springs in relation to the longitudinal ones upsets all calculations. An exaggerated reproduction of the situation that the rear springs sometimes take in a C-1-2 transmission is shown in Fig. 30. This deformation takes place in a lesser degree when one spring flattens more than another in ordinary types of transmissions with the perches fixed solid on the axle, especially when there are half or transverse springs employed as has been shown.

The combination A-B-1 is shown in Figs.

31 and 32. When the strain of the drive tends to displace the entire axle the thrust is thrown on the torque rod as well as the springs and for this reason this has been classed under the A category. When the spring deflects sufficiently powerful reactions are produced to cause the chassis to bend. The master leaf of the longitudinal springs in this type is very much exposed to rupture.

COMBINATION B-2-4—The constructor has utilized the tubular casing as the torque rod as shown in Fig. 33. This is not so dangerous as the preceding one, but it is far from good. Longitudinal efforts are not produced, but the spring tends to exfoliate and the perch should not be fixed solid.

COMBINATION C-2-4—The difficulty with this type is that it is very difficult to find a point V in Figs. 34 and 35, that will have a more or less horizontal displacement. It is possible, however, by suitably fixing the points R and S, but in sacrificing certain

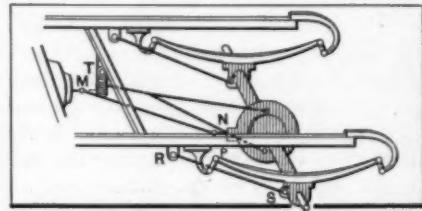


Fig. 35—Perspective view of Fig. 24, showing the method of attachment of the spring absorber to the chassis frame and the position of the distance rods and their attachments

other important considerations it is certain that the result cannot be good.

COMBINATION A-C-1-2-4 — When the springs in the displacement of the entire axle are about to deflect the T casing and the tie rods join together in preventing this, and exert thereon horizontal efforts during the period that the spring perch is endeavoring to find a position around the axis of the bridge. In such a transmission the suspension to start with is frightful, but by degrees as the different organs that perform double duties wear and take play matters improve when the parts do no work at all. But then the car is nothing better than old iron.

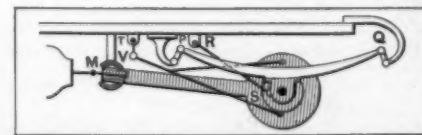


Fig. 36—The difficulty in this transmission is that there is too much of it, and when one part, such as the spring, tries to deflect it is prevented from so doing by the distance rods

If an aeroplane with a 30-foot span will lift two men, how many men will a modern big hat support?

If vice concealed is a tribute to virtue, some second-hand automobiles have many virtues.

## The Trend of Events—

### Points to the Revamping of Good Ideas and to Stability

*Granting that the hands of time point with accuracy to the things that shall come to pass, and that fallacies are the stragglers of the army of ideas, it is not too much to say that the crystallization of the automobile art cannot be brought about until the sieve of time performs its proper function and the useless pebbles are separated out, so that the keen and clever things that make for progress will shine in the lustre of their silvery coat without the stain of dross.*

- POINTS to standardization of the materials that are to be used in the makeup of automobiles.
- POINTS to harmony of relations of the materials so that each component will do its work with equal facility, and wear out in actual service at a time in common with the rest.
- POINTS to the conservation of the energies of the men in the plants so that they will do more work with less effort, accentuating quality and dignifying labor.
- POINTS to co-operation among the makers of automobiles for the good of the cause.
- POINTS to the proper regulation of production, stabilizing the industry and delivering a measure of stability to the investment on the part of the purchaser.
- POINTS to methods of production that will be automatic for quality, eliminating to a large extent the uncertainty that ordinarily greets the understanding as the "personal equation."
- POINTS to the elimination of castings on the ground that the crystals borne therein are overgrown and unruly.
- POINTS to the stopping off of the abuses that have crept into the automobile business under the appellation "heat treatment."
- POINTS to the broad idea that "doctoring" bad material to make it good as it relates to steel is a hopeless case.
- POINTS to the fundamental principle which is no better illustrated than in the process which results in the selection of suitable grades of material subjecting them to the treatment that will produce the best result.
- POINTS to the day when the users of automobiles will have less to think about with more time to enjoy life with an automobile.
- POINTS to pneumatic tires.
- POINTS to rubber.
- POINTS to sea-island cotton.
- POINTS to the sizes of tires, considering the automobiles on which they are to be used, that will accord with an investigated standard.
- POINTS to the elimination of mechanical ideas that are known among the users of automobiles for the one quality that resides within them, i. e., they are cheap.
- POINT to the time when an automobile will grow a coat of rust in the warehouse of the maker if it is loaded down with construction features that have proven their lack of competence.
- POINTS to taper fits in many of the situations that are now laboring under the handicap of insecure joints.
- POINTS to closer limits of tolerance, not only on the ground that the original cost of the automobiles would be reduced, but for the further purpose of facilitating repair work.

- POINTS to the time when automobiles will be purchased and used until they wear out in regular service, instead of being cast away when the body gets out of style.
- POINTS to the elimination of the ideas that "repeat" despite their instability and to the substitution of the things that will stay where they are put.
- POINTS to the time when the public at large will understand that the roadway is the other half of the automobile, and that the two halves must be in harmony for economy's sake.
- POINTS to the recall of the legislator who now considers it safe to dog an automobilist into jail.
- POINTS to activity on the part of automobile clubs unhampered by the politicians who now sell them out.
- POINTS to clean sport.
- POINTS to reliability tours.
- POINTS to the separation of the hampering considerations from the main situations.
- POINTS to fair dealing.

## Report of Seamless Steel Tubes Division

*Being the second report of the Seamless Steel Tubes Division of the Standards Committee of the Society of Automobile Engineers and subject to the vote of the Standards Committee.*

THE Seamless Steel Tubes Division has considered the data collected by the office of the Society as to the practice of S. A. E. members in steel tubing. A very large percentage of the sizes now in use falls within the limits of sizes included in the table given below, which is the same table that was submitted in the collection of practice data as above stated.

In addition there are many sizes distinctly individual to various automobile builders. These sizes, however, vary so much that it is impossible to include them in any table without making it so cumbersome as to be without value. This division is con-

SIZES OF COLD DRAWN SEAMLESS STEEL TUBES													OUTSIDE DIAMETER IN INCHES												
Thickness in W. O. and Fractions.	Equivalent in Decimal of Inch.	1/8	3/16	1/4	5/16	3/8	7/16	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	3	3 1/8	3 1/4	3 3/8	3 1/2	
30	.285																								
15	.240																								
10	.205																								
12	.205																								
11	.190																								
20	.184																								
8/10	.180																								
8/12	.180																								
1/8	.160																								
5/16	.155																								
3/8	.125																								
1/2	.100																								
5/8	.080																								

Table of standard cold-drawn seamless steel tube sizes recommended by the Seamless Steel Tube Division of the Standards Committee

vinced that a great many of the odd sizes in question can be very comfortably avoided by practically all of the manufacturers if they will give their attention to each particular case. Of course, there will always be certain peculiar constructions (a small percentage of the whole), wherein odd sizes must be used. This division believes that these sizes should not be included in a list of commercial standard tubing to be used in the automobile business.

If, as a result of our investigation and report, the various tube mills will carry in stock the tube sizes given in the table herewith, the purchasing agents of automobile manufacturing companies will very soon learn that they are the sizes which can be obtained most easily; which will react on the automobile designing departments even if the latter do not at first follow the standard voluntarily. We believe, however, that with this table of standard sizes before them, all designers will, wherever practicable, work to the stated sizes, and in a relatively short time

eliminate nearly all of the odd sizes which are now in use.

**VARIATIONS**—One very important thing which the automobile designer ought to bear in mind is that steel tubing is a raw product, and that it should not be expected to get it down to thousandths of an inch unless paid for proportionately. It seems to be the consensus of opinion of tube makers that tubes smaller than 2" O D will vary from .005 minus to .005 plus on both I D and O D; and tubes 2" and larger will vary from .010 minus to .010 plus on both I D and O D sizes.

We therefore recommend the adoption of the table referred to as the table of standard cold-drawn seamless steel tube sizes for automobile engineering.

(Signed) H. W. ALDEN, *Chairman.*

H. L. BARTON,  
WM. H. TUTHILL,  
JAY J. DUNN,  
H. S. WHITE,

## Worm Gear as Applied to Automobile

By E. R. Whitney, Member S. A. E.

*Paper read at the summer meeting of the Society of Automobile Engineers, at Dayton, O., June 15-17. The author discusses the merits of the worm gear in its application to the motor vehicle, fortifying his argument with the results of an efficiency test on a set of Hindley worm gears as used on a 1000-pound electric delivery wagon.*

**T**HERE seems to be a popular impression, and, I am sorry to say, among some engineers as well, that the worm gear is a device that is all right for an elevator, a mechanical motion, or for a steering device where irreversibility is desired, but that it is not to be considered seriously for the driving gear of a motor car or truck. The idea seems to be a single-thread worm having a spiral angle of 5 or 10 degrees and an efficiency of 25 to 50 per cent. We often hear questions like these: Will it coast? Will it drive backwards? Is the efficiency high enough for automobile work? Does it not wear out rapidly?

**WORM GEAR SUPERIOR IN ITS FIELD**—Contrary to the opinion expressed by writers from time to time, and especially by several writers recently, the worm gear, within its legitimate field, is superior in many respects to other forms of gearing. It has its limitations, and is not applicable to all conditions, but I will endeavor to show in this paper that for certain classes of motor-driven vehicles it is superior to other devices on the score of efficiency, durability and simplicity.

The efficiency of a worm gear is a function of the thread angle and the coefficient of friction. The coefficient of friction is a function of lubricant and the nature of the bearing surfaces.

**EFFICIENCY**—The relation of the thread angle to efficiency may be expressed as the amount of sliding between surfaces for the amount of useful work done.

Frederick A. Halsey in a work on the subject has expressed this relation very clearly, and I cannot do better than to quote his words:

"The reason why an increase of pitch, other things being equal, or, in other words, an increase of the angle of the thread, gives higher efficiency will be understood from Fig. 1. If  $AB$  be the axis of the worm and  $CD$  a line representing a thread, against which a tooth of the wheel bears, it will be seen that if the tooth bears upon the thread by a pressure  $P$ , that pressure may be resolved into two components, one of which,  $EF$ , is perpendicular, while the other,  $EG$ , is parallel to the thread

surface. The perpendicular component produces friction between the tooth and thread. The useful work done during a revolution of the thread is the product of the load  $P$  and the pitch of the worm, while the work lost in friction is the product of the perpendicular pressure  $EF$ , the coefficient of friction and the distance traversed in a revolution, which is the length of one turn of the thread. Now, if the angle of the thread be doubled, as indicated, the load  $P$  remaining the same, the new perpendicular component  $FH$  of  $P$  will be slightly reduced from the old value  $EF$ , while the length of a turn of the thread will be slightly increased. Consequently, their product and the lost work of friction per revolution will not be much changed. The useful work per revolution will, however, be doubled, because, the pitch being doubled, the distance traveled by  $P$  in one revolution will be doubled; for a given amount of useful work the amount of work lost is therefore reduced by the increase in the thread angle, and, since the tendency to heat and wear is the immediate result of the lost work, it follows that that tendency is reduced. For small angles of thread the change is very rapid, and continues, though in diminishing degree, until the angle reaches a value not far from 45 degrees, when the conditions change and the lost work increases faster than the useful work, an increase of the angle of the thread beyond that point reducing the efficiency.

"This general consideration of the subject shows the principles at the bottom of successful worm design, but a more exact examination is desirable. According to Professor Barr, the efficiency of a worm gear, the friction of the thrust bearing being neglected, is:

$$e = \frac{\tan a (1 - f \tan a)}{\tan a + f} \quad (\text{approximately})$$

in which

$e$  = efficiency.

$a$  = angle of thread (being the angle  $DFI$  of Fig. 1).

$f$  = coefficient of friction.

"This formula gives no clear indication of the manner in which the efficiency varies with the angle, and the diagram, Fig. 2, has been constructed to show this to the eye. The scale at the bottom gives the angles of the thread from 0 to 90 degrees, while the vertical scale gives the calculated efficiencies, the values of which have been obtained from the equations and plotted on the diagram. In the calculations for the diagram it is necessary to assume a value for  $f$ , and this has been taken at .05 and .025.

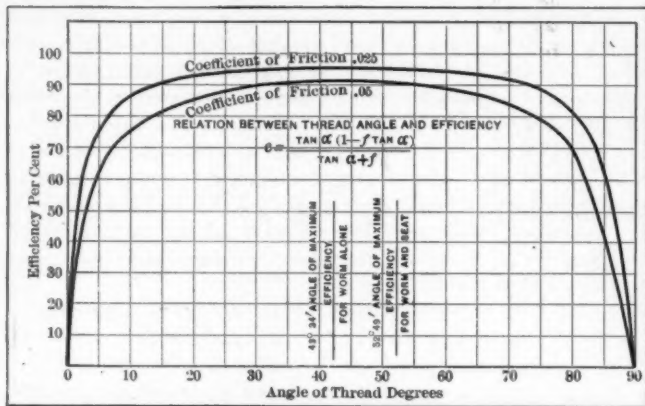


Fig. 2—Diagram constructed to show the manner in which the efficiency varies with the angle of thread

The efficiencies calculated from these values are shown in the lower and upper curves respectively. The experiments made by Mr. Wilfred Lewis for Wm. Sellers & Company showed an increase of efficiency with the speed. The present diagram may be considered as confined to a single speed, and at the same time is not to be understood as showing the exact efficiency to be expected from worms, but rather to exhibit to the eye the general law connecting the angle of the thread with the efficiency."

**TEST OF HINDLEY GEAR.**—Fig. 3 is the plotted result of an efficiency test on a set of Hindley worm gears, Fig. 4, as used in the Commercial Truck Company's 1,000-pound electric delivery wagon, Fig. 5. The data on these gears are as follows:

Pitch	.....0.96 inch
Lead	.....3.84 inches
Ratio	.....9 3/4 : 1
Center distance	.....6.796 inches
Angle of thread (average)	.....28 degrees
No. threads in worm	.....4
No. teeth in gear	.....39
Diameter of worm	.....2.8 inches
Diameter of gear	.....11.917 inches

The test was made on a stock rear construction, as illustrated in Fig. 6, the load being taken on a specially constructed Prony brake. The brake drum was mounted

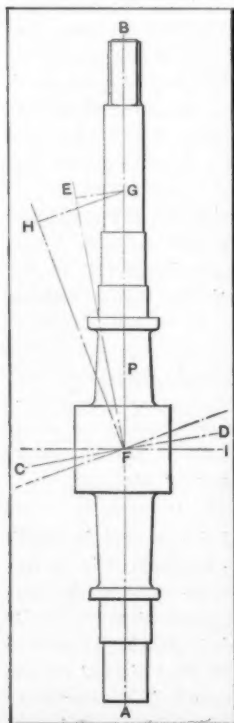


Fig. 1—Diagram of worm gear drive shaft

on a shaft with a square end passing through the square holes in both gears of the differential. The torque values were taken by a platform scale. The worm gear was driven by the electric motor, also shown in Fig. 6, which is a standard series-wound automobile type motor, and which has a normal rating of 85 volts, 22 amperes, 1200 revolutions per minute. Brake tests were first made on the motor, and before starting the test for efficiency, observations were made at various loads to determine the effect of varying the voltage and consequently the speed; it was found that the torque values on the Prony brake measuring the output of the gears were practically constant for a given current, with a considerably wide range of voltage and speed.

It is interesting at this point to note this fact in view of the results obtained by other experiments, they having found a decided change in the efficiency with changes in speed. This may possibly be accounted for by the much higher sur-

face speed of gears in this test, ranging from 200 to 800 feet per minute.

The fact that the voltage and speed could be disregarded greatly simplified the test, as it was then necessary to observe only current and torque, the efficiencies being calculated directly from the torque tests on motor and gears.

The curve of gear torque, Fig. 3, is plotted from torque values on the worm gear divided by the gear ratio. The speed curve shows the motor speed at 85 volts with varying loads.

The test corresponds to actual service conditions with the wagon and full load, working through a range of from a slight down-grade to about an 18 per cent. up-grade. The maximum efficiency of 93 per cent. corresponds to a coefficient of friction of .032.

**STEERING GEARS.**—An interesting and valuable application of the principle of worm gear efficiency may be made in connection with steering gears. The efficiency of a steering gear for trucks, and especially heavy trucks, should be as high as possible and yet with the proper degree of irreversibility, which means the proper amount of friction. I have found that these conditions can be met with worm thrusts of hardened steel, and a large spiral angle of worm thread, equally as well as with ball thrust bearings and a smaller spiral angle. The steel thrust washers are not so apt to give trouble in service and are cheaper.

The durability or life of a worm gear is a direct function of efficiency and it follows that if a gear is produced that is high enough in efficiency throughout the full load range to be practical it will also be satisfactory for durability.

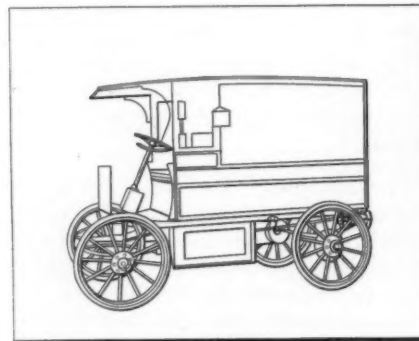


Fig. 5—Commercial Truck Company's 1,000-pound electric delivery wagon

The Commercial Truck Company's first worm-driven wagons were put into service about two years ago. Some of the first wagons have now covered approximately 25,000 miles.

Recent examination

of the gears showed very little signs of wear. I would conservatively estimate the life of these gears at from 50,000 to 60,000 miles.

**COST.**—The worm-gear drive is not a cheap device, and the results as indicated above for efficiency and durability cannot be expected unless the gears are properly designed, constructed of the best materials and accurately mounted on high-grade anti-friction bearings. Unless facilities are at hand for doing accurate machine work, the use of the worm-gear drive had better not be attempted, but once properly constructed and mounted it is practically as free from trouble as a pair of spur gears. The gears illustrated in Fig. 4 are mounted on fixed centers and with no provision for adjustment.

**THEORY AND PRACTICE.**—The Hindley worm gear is much more expensive than the straight type, but the superior bearing between the worm threads and gear teeth and the greater durability warrant the additional expense. The Commercial Hindley gear is a decidedly different product from the so-called theoretically correct Hindley gear, and in practice the writer has experienced none of the fancied difficulties in mounting accurately enough in the direction of the worm shaft to meet all practical requirements; in fact, much less accuracy is required in this direction than with the straight worm at right angles to the shaft.

There is no such thing as a so-called theoretically correct Hindley gear; that is, the threads of the worm do not have a full bearing against the whole area of all gear teeth in mesh, but the Commercial Hindley gear approaches very closely to this condition.

**RELATIVE EFFICIENCY AND FIELD.**—Some engineers claim that even in ratios that can be covered by a single reduction bevel gear the worm has advantages in a high-grade car in being much easier to make silent and has nearly equal efficiency. The efficiency is undoubtedly lower than with a single reduction bevel gear made with equal accuracy and equal mounting, but is considerably higher than a double reduction bevel and chain or chain and chain. We will assume that the legitimate field for the worm gear is only where its efficiency and durability are equal to or better than in the case of other devices to do the same work. Then, keeping in mind the principle underlying efficiency, the requirements for road clearance, etc., it would on this basis be limited to ratios of from about 6:1 to 14:1. These limits are an approximation only and are subject to modification, depending on exact conditions.

In order to put in convenient form the comparison of all features of the worm gear with other forms of gearing Table No. 1 has been made up. In consulting this table it must be understood that the comments on the different ratios are not unalterable, although they are approximately correct for the usual form of construction.

TABLE 1.—COMPARISON OF ALL FEATURES OF WORM GEARS WITH OTHER FORMS OF GEARING.

GEAR RATIO.	LOWER THAN	6:1 to 14:1	HIGHER THAN
Other forms of gearing.	6:1	Double reduction Bevel and Chain Chain and Chain	1:4
	Single reduction bevel.		Double reduction (Only required on heavy electric truck) Bevel and Chain Both chain Both spur
EFFICIENCY	LOWER	HIGHER	LOWER
DURABILITY	EQUAL OR BETTER	BETTER	EQUAL OR BETTER
NOISE	EQUAL OR LESS	LESS	LESS

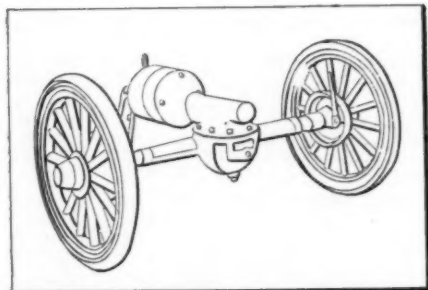


Fig. 6—Stock rear construction of the Hindley gear

**WORM GEAR IN-APPLICABLE TO GASOLINE PLEASURE CARS AND HEAVY ELECTRIC TRUCKS.**—Table No. 2 has been drawn up to give some idea of the requirements for gear ratios with different types of vehicles, and to indicate where the worm gear is applicable.

TABLE 2.—GEAR RATIO REQUIREMENTS FOR DIFFERENT TYPES OF CARS.

	Speed M.P.H.	Engine or Motor-Speed	Gear Ratio for 36" Wheels
<b>GASOLINE VEHICLES—</b>			
Pleasure car	40	1200	3.2
1000-pound wagon	20	1200	6.4
5-ton truck	10	800	8.5
<b>ELECTRIC VEHICLES—</b>			
Pleasure car	20	2000	10.7
1000-pound wagon	15	1400	10.1
5-ton truck	7	1600	24.4

It will be seen that for conditions as indicated in this table and with limitations as fixed above, that the worm gear is not applicable to gasoline pleasure cars or heavy electric trucks, but that it can be legitimately used for all gasoline business wagons and trucks, for electric pleasure cars and light electric wagons.

## Winnipeg a Motor Center

The automobile business in the Manitoban metropolis during the past year has increased nearly 150 per cent., and the astonishing influx of cars has opened the eyes of the Government to the necessity for the immediate extension of good roads.

**WINNIPEG** in 1910 had thirteen firms which were engaged in the business of selling automobiles. At the present writing, there are 31 houses carrying motor cars in stock, some of the concerns handling as many as six different types of pleasure motor cars. There are 95 different makes of machines in Manitoba alone. Of this number, 6 are electric and 67 are

gasoline machines. There are 22 models being used for commercial purposes. In 1910 there were 1,690 licenses in the Province, including dealers' demonstration motor cars. Now the licensees are divided into four classes, namely: Private owners, dealers, taxicabs and livery cars, and motorcycles. The private owners' licenses, which were

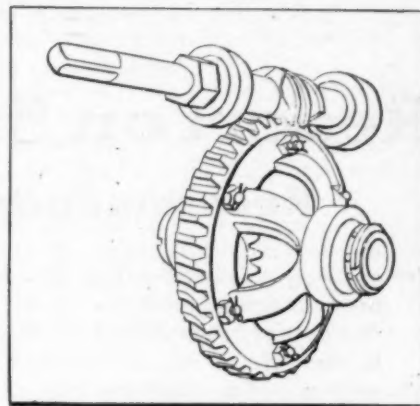


Fig. 4—Showing a set of Hindley worm gears

limited to 2,000 in the beginning, have been found inadequate in number and a new series has been issued, commencing with 3,000, the latter numerals applying to taxicabs and dealers' machines.

**RELATION OF AIR AND THE CONDUCTING ABILITY OF METAL IN RADIATORS.**—When compared with any of the metals, air is a poor conductor of heat; it follows, therefore, that any metal as used in radiator construction is capable of delivering more heat to the radiating surface than the air can take away unless the difference in temperature between the cooling air and the metallic surface is relatively great. The ability of air to absorb heat is, of course, limited to its specific heat, and to the difference in temperature between the entering and leaving air. The specific heat of air is very low, and the real problem is to have a very thin sheet of same spread out over the cooling surface. In order to increase the efficiency of the air current as it involves the cooling surface, circulation is set up; this is but another way of disregarding the poor conductivity of air for heat.

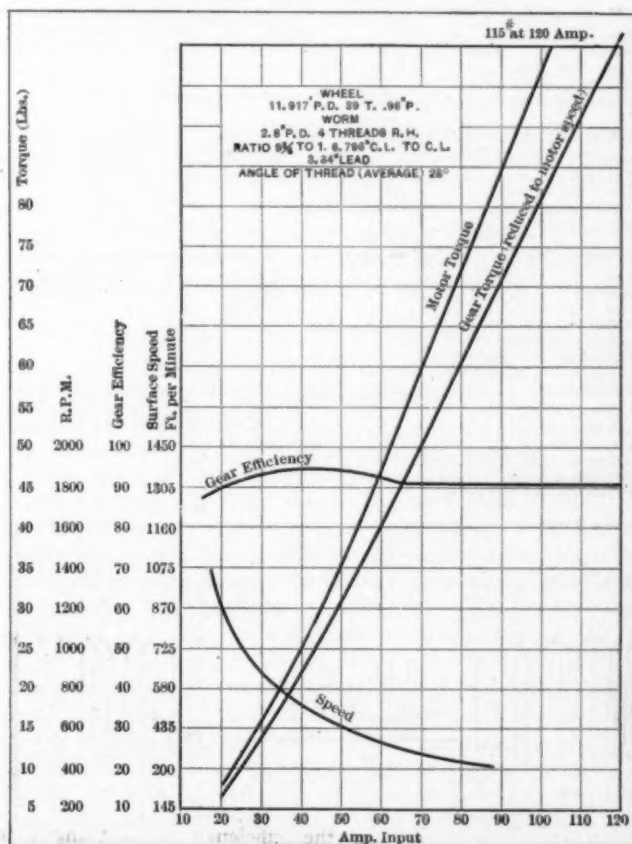


Fig. 3—Plotted result of an efficiency test on a set of Hindley worm gears, obtained from dividing the torque values by the gear ratio

# Report on Specifications for Materials

## Recommendations of Iron and Steel Division

Being the second report of the Iron and Steel Division of the Standards Committee of the Society of Automobile Engineers, as of date of May, 1911, and subject to the vote of the Standards Committee. Besides sixteen specifications for steel for various uses, with the appropriate heat treatment for each, the report recommends other standard specifications for valve metals, steel and gray iron castings and malleable iron.

**SPECIFICATIONS FOR STEEL**—These steels may be of open hearth, crucible or electric manufacture, and must be sound and free from physical defects, such as seams, heavy scale or scabs and surface defects.

These steels will be purchased on the basis of chemical analysis. The specifications indicate the desired chemical composition. Any shipments not conforming to these specifications after careful check analysis may be rejected.

**Specification No. 1.—.15 Carbon Steel**—The following composition is desired:

Carbon .....	.08% to .18%	(.15% desired)
Manganese .....	.40% to .60%	(.50% desired)
Silicon, not over.....	.20%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	

**Specification No. 2.—.20 Carbon Steel**—The following composition is desired:

Carbon .....	.15% to .25%	(.20% desired)
Manganese .....	.50% to .80%	(.65% desired)
Silicon, not over.....	.20%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	

**Specification No. 3.—.30 Carbon Steel**—The following composition is desired:

Carbon .....	.25% to .35%	(.30% desired)
Manganese .....	.50% to .80%	(.65% desired)
Silicon, not over.....	.20%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	

**Specification No. 4.—.45 Carbon Steel**—The following composition is desired:

Carbon .....	.40% to .50%	(.45% desired)
Manganese .....	.50% to .80%	(.65% desired)
Silicon, not over.....	.20%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	

**Specification No. 5.—.80 Carbon Steel**—The following composition is desired:

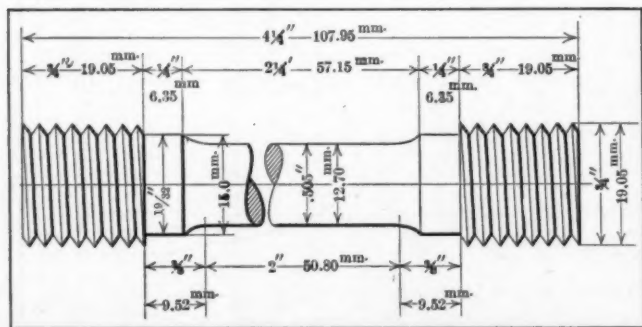
Carbon .....	.75% to .90%	(.80% desired)
Manganese .....	.25% to .50%	(.35% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.035%	
Sulphur, not over.....	.035%	

(Primarily for Springs.)

**Specification No. 6.—.95 Carbon Steel**—The following composition is desired:

Carbon .....	.90% to 1.05%	(.95% desired)
Manganese .....	.25% to .50%	(.35% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.035%	
Sulphur, not over.....	.035%	

(Primarily for Springs.)



Standard steel tensile test-specimen

**Specification No. 7.—.20 Carbon, 3½ Per Cent. Nickel Steel**—The following composition is desired:

Carbon .....	.15% to .25%	(.20% desired)
Manganese .....	.50% to .80%	(.65% desired)
Silicon, not over.....	.20%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	
Nickel .....	3.25% to 3.75%	(3.50% desired)

(Primarily for Case-Hardening.)

**Specification No. 8.—.30 Carbon, 3½ Per Cent. Nickel Steel**—The following composition is desired:

Carbon .....	.25% to .35%	(.30% desired)
Manganese .....	.50% to .80%	(.65% desired)
Silicon, not over.....	.20%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	
Nickel .....	3.25% to 3.75%	(3.50% desired)

(Primarily for Heat Treatment.)

**Specification No. 9.—.15 Carbon, Chrome Nickel Steel**—The following composition is desired:

Carbon .....	.10% to .20%	(.15% desired)
Manganese .....	.40% to .60%	(.50% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	
Nickel .....	1.50% to 2.00%	(1.75% desired)
Chromium .....	.50% to 1.00%	(.75% desired)

(Primarily for Frames.)

**Specification No. 10.—.20 Carbon, Chrome Nickel Steel**—The following composition is desired:

Carbon .....	.15% to .25%	(.20% desired)
Manganese .....	.30% to .50%	(.40% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	
Nickel .....	3.25% to 3.75%	(3.50% desired)
Chromium .....	1.25% to 1.75%	(1.50% desired)

**Specification No. 11.—.30 Carbon, Chrome Nickel Steel**—The following composition is desired:

Carbon .....	.25% to .35%	(.30% desired)
Manganese .....	.30% to .50%	(.40% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	
Nickel .....	3.25% to 3.75%	(3.50% desired)
Chromium .....	1.25% to 1.75%	(1.50% desired)

**Specification No. 12.—.45 Carbon, Chrome Nickel Steel**—The following composition is desired:

Carbon .....	.40% to .50%	(.45% desired)
Manganese .....	.40% to .60%	(.50% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	
Nickel .....	1.50% to 2.00%	(1.75% desired)
Chromium .....	.80% to 1.20%	(1.00% desired)

**Specification No. 13.—.20 Carbon, Chrome Vanadium Steel**—The following composition is desired:

Carbon .....	.15% to .25%	(.20% desired)
Manganese .....	.40% to .70%	(.50% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	
Chromium .....	.80% to 1.10%	(.90% desired)
Vanadium, not less than.....	.10%	(.18% desired)

(Primarily for Case-Hardening.)

**Specification No. 14.—.30 Carbon, Chrome Vanadium Steel**—The following composition is desired:

Carbon .....	.25% to .35%	(.30% desired)
Manganese .....	.40% to .70%	(.50% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	
Chromium .....	.80% to 1.10%	(.90% desired)
Vanadium, not less than.....	.10%	(.18% desired)

(Primarily for Heat-Treatment.)

**Specification No. 15.—.45 Carbon, Chrome Vanadium Steel**—The following composition is desired:

Carbon .....	.40% to .50%	(.45% desired)
Manganese .....	.60% to .90%	(.75% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.035%	
Sulphur, not over.....	.035%	
Chromium .....	1.00% to 1.30%	(1.20% desired)
Vanadium, not less than.....	.10%	(.18% desired)

**Specification No. 16.—Silico-Manganese Steel**—The following composition is desired:

Carbon .....	.45% to .55%	(.50% desired)
Manganese .....	.60% to .80%	(.70% desired)
Silicon .....	1.90% to 2.20%	(2.00% desired)
Phosphorus, not over.....	.04%	
Sulphur, not over.....	.04%	

NOTE—The foregoing specifications Nos. 1 to 16 are drawn to meet ordinary manufacturing conditions. Steels of greater purity can be obtained, that is, with lower limits of phosphorus and sulphur, say .025 per cent. Also narrower limits of carbon, manganese and other elements can be obtained under special terms.

**Specification No. 17.—Common Screw Stock**—The following composition covers two types of screw stock:

Carbon .....	.08% to .25%
Manganese .....	.30% to .80%
Phosphorus, not over.....	.16%
Sulphur .....	.05% to .15%

**Specification No. 18.—Low Alloy Steel**—There has lately come on the market a type of steel which sells at a slight advance over the price of open hearth and which is superior to it in a great many properties. These steels are of any carbon desired, and of the following approximate composition:

Carbon .....	As desired
Manganese .....	.50% to .80%
Silicon, not over.....	.20%
Phosphorus, not over.....	.04%
Sulphur, not over.....	.04%
Nickel .....	.90% to 1.20%
Chromium .....	.15% to .40%

#### VALVE METALS.

**Specification No. 19.—Valve Metal No. 1**—This metal shall contain not less than 26% of nickel. This material must be malleable.

**Specification No. 20.—Valve Metal No. 2**—This metal shall contain:

Carbon, not over.....	.50%
Manganese, not over.....	1.50%
Phosphorus, not over.....	.04%
Sulphur, not over.....	.06%
Nickel .....	28.00% to 35.00%

The remainder to be iron.

#### STEEL CASTINGS.

**Specification No. 21.**—The following composition is desired:

Carbon .....	.30% to .40%	(.35% desired)
Manganese .....	.60% to .80%	(.70% desired)
Silicon .....	.10% to .30%	
Phosphorus, not over.....	.06%	
Sulphur, not over.....	.06%	

#### GRAY IRON CASTINGS.

**Specification No. 22.**—The following composition is desired:

Total Carbon.....	3.25% to 3.50%	
Manganese .....	.40% to .70%	(.50% desired)
Silicon .....	1.90% to 2.20%	(2.00% desired)
Phosphorus .....	.60% to 1.00%	
Sulphur, not to exceed.....	.10%	

#### MALLEABLE IRON.

**Specification No. 23.**—The following composition is desired:

Manganese .....	.30% to .70%	(.50% desired)
Silicon, not over.....	1.00%	(.60% desired)
Phosphorus, not over.....	.20%	(.17% desired)
Sulphur, not over.....	.06%	

#### Notes and Instructions Referring to Materials Specified Hereinbefore

**STEELS**—The materials specified in detail hereinbefore include those most important to the builder of automobiles. It is obvious that there are more kinds of material specified than are likely to be used by any one manufacturer. The number given presents a choice sufficient to cover many designs. A few grades of steel properly selected and properly treated are enough to put into the parts of any given car.

**SPECIFICATION NO. 1.—15 CARBON STEEL**—The source of this material is always the basic open hearth furnace. It is a material commonly used for seamless tubing, pressed steel frames, pressed steel brake-drums, sheet steel brake-bands and pressed steel parts of many varieties. It is soft and ductile and will stand much deformation without cracking.

This steel is weak, and in an annealed condition will have an elastic limit of about 30,000 pounds per square inch, with very high reduction of area and elongation.

In the cold drawn or cold rolled condition the elastic limit may be as high as 60,000 pounds per square inch, with but little reduction of area and elongation. This relatively high elastic limit can be obtained only in light or small sections, either sheet or rod form.

This material should not be used for such parts as are to be machined. It will tear badly in the turning, threading and broaching operations.

Heat treatment produces but little benefit, and that not in strength but in toughness. It is possible to quench this grade of steel and put it in a condition to machine a little better than in the annealed state; but this is an expedient that should be used only in an emergency and not as a regular practice.

Forgings may be and are made of this steel, but it is not desirable material for forgings for the above stated reasons.

The heat treatment which will produce a little stiffness is to quench at 1500 deg. F. in oil or water. No drawing is required.

This steel will case-harden but should not be chosen for that purpose.

**SPECIFICATION NO. 2.—20 CARBON STEEL**—The natural source of supply for this material is the basic open hearth process.

This quality of steel is intended primarily for case-hardening purposes. It forges well and machines well. It may therefore be used for a very large variety of forged, machined and case-hardened parts of an automobile where strength is not paramount.

This steel is a soft, ductile material having an elastic limit in the annealed condition of about 35,000 pounds per square inch.

Steel of this quality may be also drawn into tubes and rolled into cold rolled forms, and, as a matter of fact, makes a better frame because of the slightly higher carbon and resulting strength. The increased carbon has no detrimental effect as far as usage is concerned, and it is only the most difficult cold forming operations that cause it to crack during the forming. For automobile parts it may be safely used interchangeably with Specification No. 1 as far as cold pressed shapes are concerned.

The elastic limit in a cold rolled or cold drawn condition is about 70,000 pounds per square inch; with the same degree of working as will give 60,000 pounds per square inch with Specification No. 1.

Heat treatment of this steel produces but little change as far as strength is concerned, but does cause a desirable refinement of grain after forging, and the toughness is materially increased. A simple quenching operation from about 1,500 deg. F. in oil is all that is necessary. The treatment will often help the machining qualities.

The case-hardening treatment is the most important for this quality of steel. The character of the treatment must depend upon the importance of the part to be treated and upon the shape and size. There is a certain group of parts in an automobile which are not called upon to carry much load or withstand any shock. The only requirement is hardness. Such parts are fairly illustrated by screws and by rod end pins. The simplest form of case-hardening will suffice, viz.:

**Heat Treatment No. 1**—After forging or machining—

1. Carbonize at a temperature between 1,600 deg. F. and 1,750 deg. F. (1,650 deg.-1,750 deg. F. desired).
2. Cool slowly or quench.
3. Reheat to 1,450 deg.-1,500 deg. F. and quench.

Operation 2 may be either a slow cooling or quenching, depending entirely upon the convenience and shape of the parts. If the parts are a large number of small pieces, it is very convenient to cool them quickly by quenching at a temperature somewhat below the case-hardening temperature. The subsequent heating operation (No. 3) refines the grain of the outside, carbonized steel and produces good enough results for the class of parts in question.

Another class of parts demands the best treatment, such as gears, steering-wheel pivot-pins, cam-rollers, push-rods and many similar details of an automobile which the manufacturer learns by experience must be not only hard on the exterior surface but possess strength as well. The desired treatment is one which first refines and strengthens the interior and uncarbonized metal. This is then followed by a treatment which refines the exterior, carbonized, or high-carbon, metal.

*Heat Treatment No. 2—After forging or machining—*

1. Carbonize at temperature between 1,600 deg. F. and 1,750 deg. F. (1,650-1,700 deg. F. desired).
2. Cool slowly in the carbonizing mixture.
3. Reheat to 1,500 deg.-1,550 deg. F.
4. Quench.
5. Reheat to 1,400 deg.-1,450 deg. F.
6. Quench.
7. Draw in hot oil at a temperature which may vary from 300 deg.-450 deg. F., depending upon the degree of hardness desired.

In the case of very important parts, the last drawing operation should be continued from one to three hours, to insure the full benefit of the operation.

The objects of drawing are two-fold: First, and not least important, is the relieving of all internal strains produced by quenching; second, a decrease in hardness, which is sometimes desirable. The hardness begins to decrease very materially from 350 deg. F. up, and the operation must be controlled as dictated by experience with any given part.

There are certain very important pieces that demand all of these operations, but the last drawing operation may be omitted with a large number. Experience teaches what degree of hardness and toughness combined is necessary for any given part. It is impossible to lay down a general rule covering all different uses. If the fundamental principle is well understood, there should be no trouble in developing the treatment to a proper degree.

Following the foregoing treatment a fractured part should show a fine grain on the exterior, without any appearance of shiny crystals. The smaller the crystals the better. The interior may show a silky, fibrous condition or a fine crystalline condition; but it must not show a coarse, shiny, crystalline condition.

**SPECIFICATION NO. 3—30 CARBON STEEL**—The natural source of supply for this material is the basic open hearth.

This steel is primarily for use as a structural steel. It forges well, machines well and responds to heat treatment in the matter of strength as well as toughness; that is to say, intelligent heat treatment will produce marked increase in the elastic limit. It may be used for all forgings such as axles, driving-shafts, steering pivots and other structural parts. It is the best all-round structural steel for such use as its strength warrants.

In an annealed condition the elastic limit is about 45,000 pounds per square inch, with ample elongation and reduction of area.

Heat treatment for toughening and strength is of importance with this steel. The heat treatment must be modified in accordance with the experience of the individual user, to suit the size of the part treated and the combination of strength and toughness desired. The steel should be heat treated in all cases where reliability is important.

Machining may precede the following heat treatment, depending somewhat upon convenience and the character of the treatment. If the highest strength is demanded, then strong quenching methods must be employed; for example brine. In such case, the elastic limit will be correspondingly high and the steel correspondingly hard and difficult to machine. On the other hand, if a moderately high elastic limit is all that is desired, an oil quench will suffice and machining may follow without any difficulty whatever.

*Heat Treatment No. 3—After forging or machining—*

1. Heat to 1,500 deg. F.
2. Quench.
3. Reheat to 600 deg.-1,300 deg. F. and cool slowly.

This is the simplest form of heat treatment. The drawing operation (No. 3) must be varied to suit each individual case. If great toughness and little increased strength are desired, the higher drawing temperatures may be used, that is in the neighborhood of 1,100 deg. F., 1,200 deg. F. or 1,300 deg. F. If much strength is desired and little toughness, the lower temperatures

are available. Even the lowest of the temperatures given will produce a quality of steel, after oil quenching, that is very tough—sufficiently tough for many uses. In fact, with some parts the drawing operation (No. 3) may be entirely omitted.

Results better than obtainable with the above sequence of operations may be obtained by a so-called double treatment, viz.:

*Heat Treatment No. 4—After forging or machining—*

1. Heat to 1,500 deg. F.
2. Quench.
3. Reheat to 1,400 deg. to 1,450 deg. F.
4. Quench.
5. Reheat to 600 deg. to 1,200 deg. F. and cool slowly.

This produces a refinement of grain not possible with one treatment and is resorted to in parts where extreme good qualities are desired.

Elastic limits varying from 45,000 pounds per square inch in the annealed condition, up to nearly double this amount, are obtainable by proper combinations of quenching and drawing operations. A few practical tests will teach the best method of control. High elastic limits are safe so long as they are accompanied by generous reduction of area, say, 45 per cent. or better. Such steel will withstand shock and alternate stress.

This quality of steel is not intended for case-hardening, but by careful treatment it may be safely case-hardened. This should be in emergencies only, rather than as a regular practice and, if at all, only with the double treatment followed by the drawing operation; that is, the most painstaking form of case-hardening.

**SPECIFICATION NO. 4—45 CARBON STEEL**—The natural sources of supply for this steel are various—basic or acid open hearth, crucible or electric, the most natural source being the basic open hearth.

This quality represents a structural steel of greater strength than Specification No. 3. Its uses are more limited and are confined in a general way to such parts as demand a high degree of strength and a relatively low degree of toughness. At the same time with proper heat treatment the fatigue-resisting qualities are very high—higher than with any of the foregoing specifications.

This steel is commonly used for crank-shafts, driving-shafts and propeller-shafts. It has also been used for transmission gears, but it is not quite hard enough without case-hardening and is not tough enough with case-hardening to make safe transmission gears. It should not be used for case-hardened parts, except in an emergency. Other specifications are decidedly better for this purpose.

In an annealed condition this steel should have an elastic limit of about 50,000 pounds per square inch, and in a heat-treated condition this figure may be nearly doubled.

The best heat treatment for this quality of steel for crank-shafts and similar uses is as follows:

*Heat Treatment No. 5—After forging or machining—*

1. Heat to 1,150 deg. F.
2. Quench.
3. Anneal by heating to 1,450 deg. F.
4. Cool slowly in furnace, in lime or in soft coal.
5. Reheat to 1,400 deg. to 1,500 deg. F.
6. Quench.
7. Heat to 800 deg. to 1,000 deg. F. and cool slowly.

**SPECIFICATION NO. 5—80 CARBON STEEL**—The source of this steel may be open hearth, crucible or electric furnace.

As stated under the specifications, this quality of steel is primarily for springs, and, generally speaking, for springs of light section.

The tensile test figures for this steel are unimportant inasmuch as it is intended for spring use.

The hardening and drawing of springs, that is, the heat treatment of them, is, as a rule, in the hands of the springmaker, but in case it is desired to treat, as for small coil springs, the following treatment is recommended:

*Heat Treatment No. 6—*

1. Coil.
2. Heat to 1,400 deg. to 1,450 deg. F.
3. Quench in oil.
4. Reheat to 400 deg., 500 deg. or 600 deg. F., in accordance with degree of temper desired, and cool slowly.

It must be understood that the higher the drawing temperature (Operation 4) the lower will be the elastic limit of the material. On the other hand, if the material be drawn at too low a temperature it will be brittle. A few practical trials will locate the best temper for any given use.

**SPECIFICATION No. 6—.95 CARBON STEEL**—The natural source for this steel will be the same as for Specification No. 5—it may be made in the basic open hearth, crucible or electric furnace.

This grade of spring steel is suited for the most important springs. Properly heat treated, extremely good results are possible. Substantially the same remarks apply to this quality of steel as to Specification No. 5. It is possible that the quenching temperature (Operation 2, Heat Treatment 6) of the heat treatment may be lowered slightly because of the increase in carbon, and it is also probable that the drawing temperature (Operation 4) will be at a different temperature.

Sufficient data to specify the exact physical tests of hardness or other characteristics are not yet available in the spring industry, and it is therefore impossible to draw a hard and fast set of specifications. Such data are now being accumulated.

**SPECIFICATION No. 7—.20 CARBON, 3 1-2 PER CENT. NICKEL STEEL**—The source of this steel will be the open hearth, crucible or electric furnace.

As stated under the specification, this quality of nickel steel is primarily for case-hardening. The carbon is relatively low, which means that the interior metal of a case-hardening piece will be tough.

The elastic limit of this material in an annealed condition is 45,000 pounds per square inch, with good reduction and elongation. It will respond well to heat treatment, with an increase of elastic limit up to 60,000 pounds or 70,000 pounds per square inch and with better reduction of area than in the annealed state. As before stated, this steel is not primarily for structural purposes, but for case-hardening; consequently the physical characteristics either annealed or heat treated are of minor importance. It is intended for case-hardened gears, both the bevel driving and transmission systems, and for such other case-hardened parts as demand a very tough, strong steel with a hardened exterior.

The case-hardening sequence may be varied considerably, as with Specification No. 2, those parts of relatively small importance requiring a simpler form of treatment. As a rule, however, those parts which require the use of nickel steel generally require the best type of case-hardening, viz.:

*Heat Treatment No. 7—After forging or machining—*

1. Carbonize at a temperature between 1,600 deg. F. and 1,750 deg. F. (1,650 deg. to 1,700 deg. F. desired).
2. Cool slowly in the carbonizing material.
3. Reheat to 1,450 deg. to 1,525 deg. F.
4. Quench.
5. Reheat to 1,400 deg. to 1,450 deg. F.
6. Quench.
7. Reheat to a temperature from 250 deg. to 500 deg. F. (in accordance with the necessities of the case) and cool slowly.

The second quench (Operation 6) must be conducted at the lowest possible temperature at which the material will harden. It will be found that this is sometimes as low as 1,300 deg. F.

In connection with certain uses it will be possible to omit the final drawing (Operation 7) entirely, but for parts of the highest importance this operation should be followed as a safeguard. Parts of intricate shape, comprising sudden changes of thickness, sharp corners and the like, should always be drawn, in order to relieve the internal strains.

**SPECIFICATION No. 8—.30 CARBON, 3 1-2 PER CENT. NICKEL STEEL**—The natural source for this steel will be basic open hearth, crucible or electric furnace.

This quality of steel is primarily for heat treatment for structural parts where much strength and toughness are sought, such as axles, spindles, crank-shafts, driving-shafts and transmission shafts.

In an annealed condition this steel has an elastic limit of about 55,000 pounds per square inch. Under heat treatment this may be increased anywhere up to 160,000 pounds per square inch, the ductility at this latter figure being satisfactory, reduction of area of at least 45 per cent. being obtainable. This wide variation of elastic limit is obtainable by the use of different quenching mediums—brine and oil—and the difference in drawing temperatures—from 500 deg. F. up to 1,200 deg. F.

*Heat Treatment No. 8—After forging or machining—*

1. Heat to 1,450 deg. to 1,500 deg. F.
2. Quench.
3. Heat to 600 deg. to 1,200 deg. F. and cool slowly.

A higher refinement of this treatment is:

*Heat Treatment No. 9—After forging or machining—*

1. Heat to 1,450 deg. to 1,500 deg. F.
2. Quench.
3. Reheat to 1,350 deg. to 1,400 deg. F.
4. Quench.
5. Heat to 600 deg. to 1,200 deg. F. and cool slowly.

By proper regulation and changes of quenching and drawing temperatures, a wide range of physical characteristics may be obtained. The thickness of the mass treated, the volume and temperature of the quenching medium and other details peculiar to most hardening plants, must be recognized in order to get intelligent or desirable results.

This material may be case-hardened, but it is rather high carbon for the practice of the average hardening department. The lower ranges of carbon—in the neighborhood of .25—are satisfactory, but the upper ranges—in the neighborhood of .35—approach the danger point, and steel of this carbon must be correspondingly carefully handled.

**SPECIFICATION No. 9—.15 CARBON, CHROME NICKEL STEEL**—The source of this steel will be open hearth, crucible or electric furnace.

As stated in the specification, this quality of steel is primarily intended for alloy steel frames, and for heat treatment when so used. It may be used for other purposes, such as structural parts or for case-hardened parts, by suitably modified heat treatments, but as such use is not the intent of the specification, no further discussion is given along that line.

*Heat Treatment No. 10—Heat treatment for frames after forming:*

1. Heat to 1,400 deg. to 1,450 deg. F.
2. Quench.
3. Heat to 1,000 deg. to 1,200 deg. F. and cool slowly.

The exact temperatures for operations 2 and 3 must be determined experimentally, being somewhat dependent upon the thickness of the stock treated, the temperature of the quenching medium and the design. Generally speaking, thin sections, such as frames, respond very sharply to quenching operations and must be handled with corresponding care.

**SPECIFICATION No. 10—.20 CARBON, CHROME NICKEL STEEL**—The source of this steel will be open hearth, crucible or electric furnace.

This quality of steel is intended primarily for case-hardened parts of chrome nickel steel. The treatment may be so varied as to render it possible to use this quality of steel for many structural parts.

The strength of this steel in an annealed condition is not of much importance, as this alloy, as well as others, offers no material advantage over carbon steel unless it be heat treated. The heat treatment is substantially the same sequence of operations as apply to other steels already dealt with, with suitable modifica-

tion to be determined by practical experiment. An elastic limit of 120,000 pounds per square inch is possible, with a large measure of reduction of area and elongation.

The proper quenching temperature for this grade of steel is from 1,400 deg. F. to 1,500 deg. F. The drawing temperature depends upon the elastic limit desired.

Case-hardened parts demanding this high grade of steel also demand the most careful treatment, viz.:

*Heat Treatment No. 11*—After forging or machining—

1. Carbonize at a temperature between 1,600 deg. F. and 1,750 deg. F. (1,650 deg. to 1,700 deg. F. desired).
2. Cool slowly in the carbonizing mixture.
3. Reheat to 1,400 deg. to 1,500 deg. F.
4. Quench.
5. Reheat to 1,300 deg. to 1,400 deg. F.
6. Quench.
7. Heat to 250 deg. to 500 deg. F. and cool slowly.

The heating for second quench (Operation 5) should be carried out at the lowest possible temperature at which sufficient hardness may be obtained.

**SPECIFICATION NO. 11—.30 CARBON, CHROME NICKEL STEEL**—The source of this steel will be the same as in the case of Specification No. 10.

This grade of chrome nickel steel is intended primarily for structural parts of the most important character. Parts requiring this grade of steel must be heat treated; otherwise there is no gain commensurate with the increased cost of the steel.

This quality is suitable for crank-shafts, axles, spindles, drive-shafts, transmission shafts and, in fact, the most important structural parts of the highest-priced cars.

The elastic limit of the annealed material is of no importance, as this steel should not be used in an annealed state. The elastic limit of the heat treated material may be carried as high as 175,000 pounds per square inch, with generous reduction of area and elongation.

Heat treatment recommended:

*Heat Treatment No. 12*—After forging or machining—

1. Heat to 1,450 deg. to 1,500 deg. F.
2. Quench.
3. Reheat to a temperature between 500 deg. and 1,250 deg. F. and cool slowly.

A higher refinement of this same treatment is:

*Heat Treatment No. 13*—After forging or machining—

1. Heat to 1,450 deg. to 1,500 deg. F.
2. Quench.
3. Reheat to 1,400 deg. F.
4. Quench.
5. Reheat to a temperature between 500 deg. F. and 1,250 deg. F. and cool slowly.

This grade of steel is sensitive and must be handled with a great deal of care. The temperatures should be controlled by pyrometer. The lower the temperature at which the proper response to treatment is obtained, the better will be the results. At the same time, if sufficient temperature is not used, there will be an incomplete or unsatisfactory response.

This steel is not intended for case-hardening, but may be so treated in an emergency. If case-hardening is attempted, the highest degree of care must be exercised.

**SPECIFICATION NO. 12—.45 CARBON, CHROME NICKEL STEEL**—The source of this steel will be the basic open hearth, crucible or electric furnace, probably one of the last two.

The use of this steel is mostly for gears where extreme strength is necessary. The carbon is sufficiently high to cause the material, in the presence of chromium and nickel, to become sufficiently hard to make a good gear when quenched, without case-hardening (carbonizing).

The characteristics of this steel in an annealed condition are unimportant, as it should not be used in that condition. Heat treatment produces an elastic limit that may be carried to over

200,000 pounds per square inch, with good reduction of area and elongation.

This steel is difficult to forge. During the forging operation it should be kept at a thoroughly plastic heat and not hammered or worked after dropping to ordinary forging temperatures, as cracking is liable to follow. The steel also becomes so very hard as to forge with great difficulty. On the other hand, too high a temperature is not advisable, as the steel becomes red-short and breaks. In brief, the forging temperature limits are narrow, and this steel must be reheated more frequently than any of the other qualities dealt with. Generally speaking, the higher the carbon of chrome nickel steel, the more is this characteristic found to be true.

To heat treat for gears:

*Heat Treatment No. 14*—After forging—

1. Heat to 1,500 deg. F. (plus or minus 25 deg. F.).
2. Quench.
3. Reheat to 1,400 deg. to 1,450 deg. F. (Hold at this temperature one-half hour, to insure thorough heating.)
4. Cool slowly.
5. Reheat to 1,450 deg. to 1,500 deg. F.
6. Quench.
7. Reheat to 250 deg. to 550 deg. F. and cool slowly.

This steel cannot be machined unless thoroughly annealed (Operations 3 and 4).

The final drawing operation must be conducted at a heat which will produce the proper degree of hardness. The desired Brinell hardness for a gear is between 430 and 470, the corresponding Shore hardness being from 75 to 85.

This quality of steel should not be case-hardened.

**SPECIFICATION NO. 13—.20 CARBON, CHROME VANADIUM STEEL**—The source of this steel will be open hearth, crucible or electric furnace.

As noted in the specification, it is primarily for case-hardening. The uses are for the best case-hardened parts of high-priced cars; that is, case-hardened shafts, gears and like important parts.

The physical characteristics of this steel annealed are relatively unimportant. The material will respond somewhat to heat treatment and be considerably toughened thereby. Its proper use is for case-hardening, in accordance with the following treatment:

*Heat Treatment No. 15*—After forging or machining—

1. Carbonize at a temperature between 1,600 deg. F. and 1,750 deg. F. (1,650 deg. to 1,700 deg. F. desired).
2. Cool slowly in the carbonizing mixture.
3. Reheat to 1,600 deg. to 1,700 deg. F.
4. Quench.
5. Reheat to 1,300 deg. to 1,400 deg. F.
6. Quench.
7. Reheat to 250 deg. to 500 deg. F. and cool slowly.

The high initial quenching temperature of this steel is noteworthy, that is, something a little over 1,600 deg. F. This is different from the other steels referred to and characteristic of chrome vanadium steel.

The heating for second quench (Operation 5) should be conducted at the lowest possible heat that will harden the exterior, carbonized surface. Practical experiment will develop the best temperature for local conditions in any hardening room.

**SPECIFICATION NO. 14—.30 CARBON, CHROME VANADIUM STEEL**—The source of this material may be open hearth, crucible or electric furnace.

The uses for this steel are for structural purposes, very much as set forth for .30 carbon, nickel steel and for .30 carbon, chrome nickel steel that is, the important structural portions of an automobile—the crankshaft, drive-shafts, axles and the like.

The physical characteristics in an annealed condition are unimportant. The steel should not be used in that condition—not

that it is unsafe, but because there will be no gain commensurate with increased cost of the material.

For heat treatment:

*Heat Treatment No. 16*—After forging or machining—

1. Heat to 1,600 deg. to 1,700 deg. F.
2. Quench.
3. Reheat to some temperature between 500 deg. F. and 1,300 deg. F. and cool slowly.

The elastic limit obtainable after heat treatment may be between 60,000 pounds per square inch and 150,000 pounds per square inch, with good toughness as represented by reduction of area and elongation.

This steel may be case-hardened, but if so treated it must be handled with care on account of the relatively high carbon.

**SPECIFICATION NO. 15—45 CARBON, CHROME VANADIUM STEEL**—The source of this steel may be open hearth, crucible or electric furnace.

This quality of steel contains sufficient carbon in a combination with vanadium to harden when quenched at a proper temperature.

The elastic limit after suitable heat treatment may be carried up to the neighborhood of 200,000 pounds per square inch, with reduction of area great enough to indicate good toughness.

This steel may be used for structural parts where exceedingly great strength is required.

The heat treatment may be as described for the .30 carbon, chrome vanadium steel; and the drawing temperature must be suitably modified to produce proper toughness. For gears this steel should be annealed after forging, the treatment to be about as follows:

*Heat Treatment No. 17*—

1. Heat to 1,600 deg. F.
2. Quench.
3. Reheat to 1,450 deg. F.
4. Cool slowly.
5. Reheat to 1,600 deg. to 1,650 deg. F.
6. Quench.
7. Reheat to 350 deg. to 550 deg. F. and cool slowly.

This last drawing operation must be modified to obtain any desired hardness.

**SPECIFICATION NO. 16—SILICO-MANGANESE STEEL**—The source of this steel will be the open hearth, crucible or electric furnace.

This steel is intended for structural parts, springs and gears. It should not be used without heat treatment.

The physical characteristics in the heat treated condition are similar to those of other alloy steels specified, the elastic limit being under control and ranging from 60,000 pounds per square inch to 175,000 pounds per square inch, with good reduction of area and elongation.

For structural parts the treatment should be:

*Heat Treatment No. 18*—After forging or machining—

1. Heat to 1,650 deg. to 1,750 deg. F.
2. Quench.
3. Reheat to a temperature between 600 deg. F. and 1,400 deg. F. and cool slowly.

Suitable temperatures for any given thickness of piece and the character of the quenching medium must be determined experimentally.

When used for springs, this material may be treated as above with proper modification as to drawing temperature, with the probability that 800 deg. F. as a drawing temperature will give about the proper characteristics.

**SPECIFICATION NO. 17—COMMON SCREW STOCK**—This steel may be made by any process.

There are two types of screw stock commonly found in the market. The controlling element in one type is phosphorus, which is commonly found between .10 and .20 per cent. The controlling element in the other type is sulphur, which is commonly found between .07 and .15 per cent.

Ordinary screw stock is a free machining and very cheap steel, lacking strength and toughness. It is an unsafe steel to be used in the vitals of an automobile.

Screws of these materials from hot-rolled bars should be heat treated and not used in a rolled or annealed condition. Screws made from cold-rolled bars are much stronger than if in a rolled or annealed condition. But the best results from both types of steels may be obtained if heat treated.

Heat Treatment No. 3 is suitable for screws. Heat treatment after machining produces the strongest screw. If machined after treatment the grain of the material is, in effect, nicked by the thread and thereby weakened.

**SPECIFICATION NO. 18—LOW ALLOY STEEL**—This specification represents a class of steels that are made from ores containing chromium and nickel. Steels made from such ores may be referred to as natural alloy steels. At the same time other steels closely duplicating them in analysis are made by the addition of chromium- and nickel-containing materials to an otherwise simple steel.

These natural alloy steels are low in cost as compared with chrome-nickel steels in general. This is to be expected in view of the fact that the alloying elements are low in percentage and cost little in view of their origin. These steels are not intended to compete with the higher alloys of chromium or nickel, but are intended to furnish a better quality than carbon steel at a very slight increase in cost.

As found in the market, these steels contain about 1.00 per cent. of nickel and from .15 to .40 per cent. of chromium. Of such composition, there is a material gain in strength and toughness as the result of heat treatment.

These steels are obtainable in any desired carbon from .10 to 1.00 per cent. At a slight increase in cost these steels may be obtained containing from .30 to .60 per cent. chromium, which composition results in a very decided gain in physical characteristics.

The natural alloy steels respond to case-hardening treatments better than the plain carbon steels and the nickel steels of similar carbon content. For example, a .20 carbon, natural alloy steel makes an excellent gear when properly carbonized and treated in accordance with Heat Treatment No. 7.

For structural purposes the .30 carbon or even the .40 carbon steel may be used, properly treated, in accordance with Heat Treatment No. 8 or No. 9.

**SPECIFICATIONS NOS. 19 AND 20—VALVE METALS**—These materials are high-nickel valve metals. They do not respond to heat treatment. The best that can be done with them is to treat for the purpose of securing uniformity of condition, by annealing at ordinary temperatures or by quenching from ordinary temperatures (1,500 deg. F. or thereabouts). Change of strength or ductility cannot be expected to any commercial degree.

**SPECIFICATION NO. 21—STEEL CASTINGS**—The specifications given for steel castings represents a quality commonly made by open hearth and crucible manufacturers.

Genuine steel castings, and not malleable iron and complex mixtures often found in the market masquerading under the name of steel, are referred to.

Genuine steel castings may be annealed or heat treated to great advantage for important parts. A steel casting of the composition given in the specification should be tough, so as to bend to a considerable angle before breaking. The elastic limit of such a casting in an annealed condition is in the neighborhood of 35,000 pounds per square inch.

The tensile strength of genuine malleable iron castings properly annealed is in the neighborhood of 30,000 pounds per square inch, with no elastic limit worth mentioning.

Like other castings, steel castings are subject to blow-holes. Consequently, they should not be used for the vital parts of an automobile. It is impossible to inspect against blow-holes. Steel castings for axles, crank-shafts and steering-spindles are used only at great risk.

**SPECIFICATION NO. 22—GRAY IRON CASTINGS**—The use of specifications for cast iron in the present state of the foundry art is not very easy. The foundryman, if he is not accustomed to work to analysis, will object, although his iron may be within the specifications given 90 per cent. of the time. Moreover, if there are any defective cylinders he will be likely to lay it to the composition of the iron, whereas the fault may lie in his foundry methods, apart from composition.

Consequently these specifications should be used as indicating the ideal mixture—something for the foundryman to work to, even though he may not be willing to guarantee the analysis.

If trouble is experienced with cylinders, analyses of samples of the iron will show whether or not the composition is somewhere near what it should be. If the composition is very far from the specification here given, the purchaser will be justified in putting up strenuous objection.

Iron in accordance with this specification will be strong and reasonably close-grained in the thicknesses cast, and one that wears well.

**SPECIFICATION NO. 23—MALLEABLE IRON**—The remarks made in connection with the gray iron specification (No. 22) apply even more strongly to malleable iron. Iron of the composition given, properly annealed, will make a strong and tough casting; but improperly annealed it will not make a good casting.

Castings that are received brittle may be so from two causes: first, unsuitable mixture of iron; second, incomplete annealing. Consequently, if brittle castings are received, they should be analyzed, and if the analysis is correct, then it is certain that the annealing operation was not properly performed.

Castings varying seriously in composition from that given in the specification, especially as to phosphorus, are liable to be brittle, even if properly annealed, and should therefore be avoided.

### List of Heat Treatments

#### Heat Treatment No. 1.

- After forging or machining—
1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
  2. Cool slowly or quench.
  3. Reheat to 1450°-1500° F. and quench.

#### Heat Treatment No. 2.

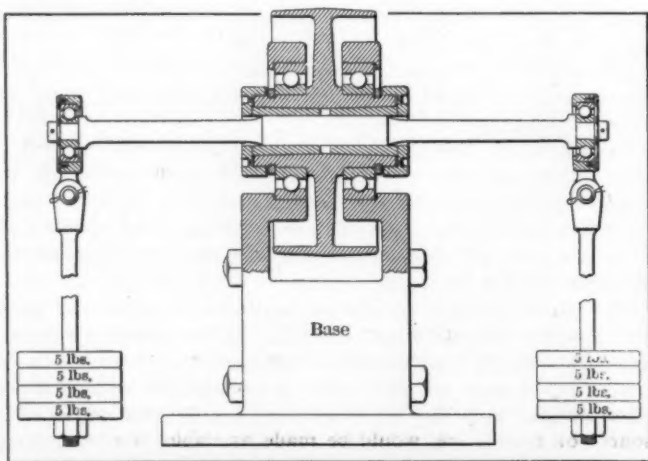
- After forging or machining—
1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
  2. Cool slowly in the carbonizing mixture.
  3. Reheat to 1500°-1550° F.
  4. Quench.
  5. Reheat to 1400°-1450° F.
  6. Quench.
  7. Draw in hot oil at a temperature which may vary from 300°-450° F., depending upon the degree of hardness desired.

#### Heat Treatment No. 3.

- After forging or machining—
1. Heat to 1500° F.
  2. Quench.
  3. Reheat to 600°-1300° F. and cool slowly.

#### Heat Treatment No. 4.

- After forging or machining—
1. Heat to 1500° F.
  2. Quench.



Cross-section of an endurance testing machine

3. Reheat to 1400°-1450° F.
4. Quench.
5. Reheat to 600°-1200° F. and cool slowly.

#### Heat Treatment No. 5.

After forging or machining—

1. Heat to 1550° F.
2. Quench.
3. Anneal by heating to 1450° F.
4. Cool slowly in furnace, in lime or in soft coal.
5. Reheat to 1400°-1500° F.
6. Quench.
7. Heat to 800°-1000° F. and cool slowly.

#### Heat Treatment No. 6.

1. Coil.
2. Heat to 1400°-1450° F.
3. Quench in oil.
4. Reheat to 400°, 500° or 600° F., in accordance with degree of temper desired, and cool slowly.

#### Heat Treatment No. 7.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing material.
3. Reheat to 1450°-1525° F.
4. Quench.
5. Reheat to 1400°-1450° F.
6. Quench.
7. Reheat to a temperature from 250°-500° F. (in accordance with the necessities of the case) and cool slowly.

#### Heat Treatment No. 8.

After forging or machining—

1. Heat to 1450°-1500° F.
2. Quench.
3. Heat to 600°-1200° F. and cool slowly.

#### Heat Treatment No. 9.

After forging or machining—

1. Heat to 1450°-1500° F.
2. Quench.
3. Reheat to 1350°-1400° F.
4. Quench.
5. Heat to 600°-1200° F. and cool slowly.

#### Heat Treatment No. 10.

1. Heat to 1400°-1450° F.
2. Quench.
3. Heat to 1000°-1200° F. and cool slowly.

#### Heat Treatment No. 11.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing mixture.
3. Reheat to 1400°-1500° F.
4. Quench.
5. Reheat to 1300°-1400° F.
6. Quench.
7. Heat to 250°-500° F. and cool slowly.

#### Heat Treatment No. 12.

After forging or machining—

1. Heat to 1450°-1500° F.
2. Quench.
3. Reheat to a temperature between 500° F. and 1250° F. and cool slowly.

#### Heat Treatment No. 13.

After forging or machining—

1. Heat to 1450°-1500° F.
2. Quench.
3. Reheat to 1400° F.
4. Quench.
5. Reheat to a temperature between 500° F. and 1250° F. and cool slowly.

#### Heat Treatment No. 14.

After forging—

1. Heat to 1500° F. (plus or minus 25° F.)
2. Quench.
3. Reheat to 1400°-1450° F. (Hold at this temperature one-half hour, to insure thorough heating.)
4. Cool slowly.
5. Reheat to 1450°-1500° F.
6. Quench.
7. Reheat to 250°-550° F. and cool slowly.

#### Heat Treatment No. 15.

After forging or machining—

1. Carbonize at a temperature between 1600° F. and 1750° F. (1650°-1700° F. desired).
2. Cool slowly in the carbonizing mixture.
3. Reheat to 1600°-1700° F.
4. Quench.
5. Reheat to 1300°-1400° F.
6. Quench.
7. Reheat to 250°-550° F. and cool slowly.

#### Heat Treatment No. 16.

After forging or machining—

1. Heat to 1600°-1700° F.
2. Quench.
3. Reheat to some temperature between 500° F. and 1300° F. and cool slowly.

#### Heat Treatment No. 17.

After forging—

1. Heat to 1600° F.
2. Quench.
3. Reheat to 1450° F.
4. Cool slowly.
5. Reheat to 1600°-1650° F.
6. Quench.
7. Reheat to 350°-550° F. and cool slowly.

## Heat Treatment No. 18.

After forging or machining—

1. Heat to 1650°-1750° F.
2. Quench.
3. Reheat to a temperature between 600° F. and 1400° F. and cool slowly.

## Definitions and Notes

**TENSILE STRENGTH** per square inch indicates the strength the steel would have tested in a rod of 1-inch area cross-section. This standard is adopted so that no matter what the size of the test-specimen may be, the results are calculated to a square inch basis.

**ELASTIC LIMIT**—The figure generally referred to in engineering practice as "elastic limit" is not the true elastic limit, except in those instances where close reading extensometers are used. The elastic limit usually referred to is that obtained by either the drop of the beam of the testing machine or by means of dividers which indicate when the test-specimen begins to stretch visibly as noted by the aid of dividers. Elastic limit is that point at which, if the load on the steel specimen be released, the specimen will return to its original length before being placed under tension test, and beyond which, if the load be released, the specimen will not return to its original length. The elastic limit is really the point necessary to use in figuring all strains and stresses for structural purposes.

**REDUCTION OF AREA** is the measured amount (in per cent.) that a test-specimen reduces in size at the point of fracture. This element is valuable because it shows the condition of the steel in the specimens. For instance, a coarse-grained, overheated steel, or an under-worked steel, shows a low percentage of reduction of area and is, therefore, to be avoided. On the other hand, a steel well hammered or rolled, or heat treated, shows a high percentage of reduction of area as the result of a very fine grain.

**ELONGATION** represents the ductility and softness of the material. A considerable degree of elongation is required in order that there may be nothing abnormal about the material being tested. It is of the least importance for automobile purposes.

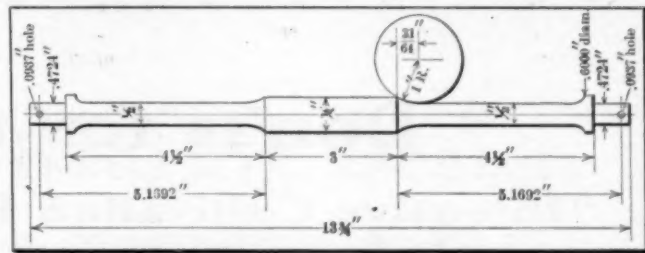
**ENDURANCE**—In the foregoing specifications "endurance" has been referred to. Endurance is conveniently studied by means of a so-called alternate stress or endurance machine. A cut of the form of test-specimen required and the machine in which it is tested are shown herewith. The machine is of standard design and can be obtained in the market from builders of testing machines.

It is a fact that tensile tests giving elastic limit, maximum strength, reduction of area and elongation, indicate the probable endurance of a steel. At the same time, the endurance of a steel is not always proportional to the increase of elastic limit or increase of elongation and reduction of area. Consequently, the endurance machine is the best method of studying the real effect of heat treatments and of material. It is not as commercial as the tension test, as it takes more time to perform. It is, however, an excellent device to use in conjunction with the tension test.

The machine should be firmly mounted upon a solid base in order to eliminate the vibration which is sure to result from a more or less flexible base and which sets up vibration and increases or otherwise renders uncertain the effect of alternate stress within the metal itself.

**NATURAL CONDITION**, that is to say, steel in the natural condition. This means little. It is supposed to represent the condition of steel after ordinary hot rolling or forging operations. Steel in such condition may be either good or bad, depending largely upon the accidental sequence of operations and the method of carrying them out. This condition might well be called an accidental rather than a natural condition. It is not a good term to use and should be avoided. Neither should steel in a natural condition be used for any but the unimportant parts of an automobile.

Steel in a natural condition after annealing is uniform and safe, even if not as strong or as tough as after suitable heat treatment.



Endurance machine test-specimen

**ANNEALING** consists of heating to a temperature above the critical point and cooling slowly.

**DRAWING** consists of heating to some point below the critical point and cooling slowly.

**HEAT TREATING**—A sequence of heating, quenching and slow cooling operations.

**QUENCHING**—To cool suddenly in any liquid or in air.

**QUENCHING MEDIUMS**—Brine, water, oil, and many others. Brine produces the sharpest results, but is severe and has a greater tendency to warp parts, producing a higher elastic limit.

Water is intermediate between brine and oil and often may be used to advantage with parts of suitable section and thickness. Its tendency to cause warping is not as great as that of brine, nor can so high an elastic limit be obtained as with brine.

Oil—fish, cottonseed or lard—is best for quenching purposes. It is the mildest of the quenching mediums and should be used wherever it is found possible to obtain the desired results by its use.

**ALL REFERENCES TO "QUENCH" IN THE FOREGOING HEAT TREATMENTS MEAN OIL UNLESS OTHERWISE STATED.** If found by experiment that oil will not produce the desired results, then water should be tried, or even brine. Nearly all case-hardened parts may be successfully hardened in oil. Oil exerts less tendency to warp than any of the other quenching mediums mentioned.

HENRY SOUTHER, *Chairman.*

E. F. RUSSELL,  
S. V. HUNNINGS,  
A. R. GORMULLY,  
ARTHUR HOLMES,  
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THOMAS TOWNE,  
F. D. CARNEY,  
W. P. BARBA,  
JOSEPH SCHAEFFERS,  
COKER F. CLARKSON, *Secretary.*

## Reciprocity Treaty will Aid Freight Automobile

*The signing of the proposed reciprocity treaty with Canada would witness a quickening of the freight automobile trade between the two countries. The Dominion, with its comparative lack of railroad facilities, would be immensely benefited by the wide adoption of the mechanically-propelled business wagon.*

**SUCCESSFUL** negotiations of the proposed reciprocity treaty would open up an entirely new vista for the freight carrying automobile. So far the Canadian trade has been almost exclusively confined to the passenger cars, with here and there a notable exception, but in the event of reciprocity the market for the trucks would be immediately enlarged.

Precisely the same arguments in favor of the delivery truck in the cities and thickly populated areas as have been given with such convincing force to the business men of the United States will apply with equal strength to Canadian conditions. In addition, the mission of the freight automobile in handling all sorts of heavy draft work, in transporting the fruit, of the field and forest to the railroads, is of tremendous importance to Canada.

Immense territories, now considered of doubtful value as a source of agriculture, would be made available by means of the freight automobile.

# Don'ts for the Engineer

## "Intensive Cultivation Produces a Bumper Crop"

*In view of the dual relation of the "engineer," he being the man whom the maker relies upon for sound mechanical advice, and the embodiment of earnestness and unswerving integrity from the user's point of view, his acquaintance with the scenery in the borderland of chicanery should be on the same plane as a child's understanding of the intricacies of a chronometer.*

- Don't allow chance to tempt you to be satisfied with a vague statement of the position that will fall to your lot in a negotiation.
- Don't ape the taking ways of a ward politician.
- Don't specify the types of materials that are to be used in an automobile if the purchasing agent is going to base his action upon the wiles of the lowest bidder.
- Don't allow a draftsman to dress up an incongruity and put it in the working drawings of a promising automobile.
- Don't permit your ambition to run away with you to the extent of crowding a \$5,000 design into a \$1,000 automobile.
- Don't outline a big plan and neglect the little things.
- Don't emulate the fellow with the bath-towel spine when you go into conference if there is any disposition on the part of Mr. Commercialman to advocate a plain deception.
- Don't take any stock at all in the ability of the publicity department to fix quality in a car if it is not of your making.
- Don't fail to take an active part in the co-operative work that is going on in the Society.
- Don't participate in the deliberation of the council of the wise excepting to add something of value that will be in harmony with the whole.
- Don't go in with the impression that an amicable relation will be destroyed if you cannot agree to a doctrine that may be put upon the tapis for discussion; if you differ with the doctors present a reason for the disagreement, but if the majority fails to see the point it is reasonable to accept the rule of so much wisdom in preference to the opinion of a single man.
- Don't bother with poignant details; watch the main chance every minute of the time.
- Don't pre-empt the prerogatives of a disorderly minority; a good society has one leader only.
- Don't bolt the convention if you discover that some one has been nibbling upon the piece of pie that you think belongs to you.
- Don't be overawed if the flow of wisdom emanating from some unexpected quarter gets beyond your comprehension.
- Don't let the world say that you failed to hoe your row.
- Don't permit weeds to grow in your part of the garden, busying yourself in the meantime with the troubles that have been assigned to your neighbors.
- Don't be afraid of doing too much work in the good cause; the master of the vineyard will feel disposed to assign another job if you complete the task that merits the attention of the moment.
- Don't let moss grow on the back seats in the place where men assemble for the good of the cause; the curse of too many leaders should be obvious to you.
- Don't lean backwards on the staff of respectful diffidence; you might be unconscious to the "call" when it comes.

- Don't enter into any intrigue whatsoever; the position that you might acquire under such circumstances will be built upon soggy foundation.
- Don't associate with any more golden ideas than you can tote for a mile.
- Don't wear blinds—they were originally designed for horses.
- Don't look through the wrong end of the telescope—it makes a planet look like a marble.
- Don't place too much store upon what you may think as compared with what you do.
- Don't get yourself into the state of mind wherein you brave the idea that your last fond effort spells finality.
- Don't accept the rulings of a foreshortened perspective; the world will go on and leave you far behind in that event.
- Don't lay too much stress upon the value in new work of the jigs and fixtures that are a true part of last year's mistakes.
- Don't court success with a bludgeon.
- Don't try to get a society to limit its activities, crystallizing the things that you know about; your interest may be furthered by exploring the field beyond the range of your personal knowledge.
- Don't be afraid to dribble out a few of your good ideas in the common cause; you can scarcely afford to be a welsher.
- Don't try to exchange a fallacy for a truth; your debt can only be liquidated by the delivery of an equality of value.

## Small Cars Wanted in France

*In certain sections of that country there is a strong demand for a serviceable three- or four-wheeler of moderate price that, in addition to meeting the severe service demands of the French, shall, in view of the high price of fuel, be a moderate consumer of gasoline.*

THERE is a demand in the vicinity of Bordeaux, France, for moderate-priced, medium-power, small automobiles, and the manufacturer who goes about the business properly is liable to get some trade. The call is for small four-wheel or three-wheel vehicles which could be bought for \$350 or \$400, for commercial delivery purposes. There are people who are anxious for the advent of automobiles of this description. But there are certain rules to be observed, if the foreign manufacturer would sell his product in this market. The machine must be fully guaranteed to equal a French-made car; it must be qualified to endure the rigid tests according to French standardization; it must manifest its capability and its reliability by actual results, after severe demonstrations have been made in France; it must show its strength to a point of carrying 400 pounds of passengers and at the same time it must not drink up more than a gallon of gasoline to every thirty or forty miles of road traveled, gasoline being very dear in France and therefore entering considerably into the expense of keeping up a machine; the tires must conform to Continental European standards. A complete line of spare parts and accessories would need to be kept on hand at headquarters in Paris as well as in the subdepots in large provincial cities. It would be necessary to print descriptions of standards, etc., and also all catalogues in the French tongue, while a staff of American-French or French salesmen would be indispensable.

# Ball and Roller Bearing Design

By Arnold C. Koenig, Member S. A. E.

*Extracts from a paper read at the Midsummer Meeting of the Society of Automobile Engineers at Dayton, Ohio, June 15-17. The author analytically considers the stresses imposed on ball and roller bearings with special reference to automobile practice, and shows by illustrations and formulae the limits beyond which it is not safe to go in designing bearings suitable for that work. He reinforces his argument with the details of a series of tests conducted by Stribeck and other specialists along this line of mechanical research, and incidentally points out a few fallacies which obtain in present practice.*

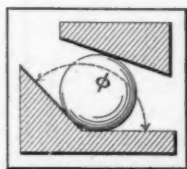


Fig. 3 — Showing frictional resistance to ball rotation in a three-point bearing

A STEEL girder may be designed to sustain a load of many tons without deflecting more than 1-500 of its length under such load, yet a load of even one ounce will produce a slight, although imperceptible, deflection.

With this fact in mind, there is another illustration which may be mentioned as introductory to the subject in hand: viz., that any two or more materials in contact under pressure will deform or give way to each other in proportion to their relative degrees of hardness or resiliency. Thus, an ordinary pneumatic tire makes no perceptible impression on a hard pavement, but leaves a well-defined track in soft yielding earth.

The wheel of an ordinary wagon, heavily loaded, passing over an asphalt pavement in summer when the asphalt is softened by the heat of the sun, will sink into the pavement very perceptibly, and the rolling action of the wheel, as the wagon moves forward, causes a small "wave" of asphalt to be pushed ahead of the wheel, as shown in Fig. 1. The theoretic effect of this "wave" on the dynamic pull required to move the load, needs no discussion here, as its practical effect is amply demonstrated by the fact that a team of horses will pull a loaded dray up a steep grade on a street paved with hard granite blocks, yet get "stalled" with the same load on a street having the same percentage of grade, but paved with asphalt.

Steel will stand reasonable stresses (within the limits of its elastic resilience) almost indefinitely, but under excessive strains oft repeated or of long duration it will slowly deteriorate under the law of the "fatigue of metals" and will eventually fail under a stress or load that may be small as compared with its original capacity.

According to Professor Merriman, the first systematic experiments upon the effect of repeated loading of elastic materials were conducted by Wöhler (1859 to 1870). He found, for example, that a bar of wrought iron subjected to tensile stress varying from zero to the maximum was ruptured by:

800 repetitions from 0 to 52,800 lbs. per sq. in.					
107,000	"	"	0 to 48,000	"	"
450,000	"	"	0 to 39,000	"	"
10,140,000	"	"	0 to 35,000	"	"

"It was found that the stress could be varied from zero up to something less than the elastic limit an indefinite number of times (several millions) before rupture occurred, but with complete reversal of stress or alternate equal and opposite stresses (tension and compression) it could be broken by a sufficient

number of applications when the maximum stress was only about one-half to two-thirds the stress at the elastic limit."

The elastic resilience of any material, as, for instance, steel, is the energy expended in producing a load deformation (or strain) at the true elastic limit of the material. If the limit of proportionality between stress and strain (within the definite range of elastic resilience) be exceeded, there results a permanent deformation or "set," and the elastic resilience for subsequent loadings is increased to the amount of energy required to produce such load deformation, but the difference between the increased elastic resilience and the ultimate load (or total load deformation at rupture) is proportionately decreased.

The maximum unit load deformation in a ball bearing must therefore be kept, as nearly as practicable, within the primary elastic resilience of the materials used.

The maximum allowable (safe) stress within the elastic resilience of any material such as steel is determined by the character of the load and the frequency with which it is applied. Thus, for a *live* load (one that is suddenly applied and suddenly released without appreciable impact velocity) the elastic resilience should be taken at one-half of that which obtains under a gradual or static load. For loads which are subject to shocks, impact or reversals of stress this should be again divided by two. The frequency of load repetitions at any point in a ball race is equal to the number of revolutions  $\times$  the number of balls  $\times$  the ratio of diameters of ball and race, or:

$$\text{Load frequency} = \text{r. p. m.} \times \text{No. of balls} \times \frac{\text{ball diam.}}{\text{race diam.}}$$

However, as stated before, the exact relation which the frequency of load repetitions bears to the safe load capacity of a bearing has thus far been impossible to determine mathematically; therefore, to the safe working value of the elastic resilience (one-half or one-fourth of elastic limit) as determined by the character of the load there should be applied a further factor of safety of four or (preferably) five to provide for variable and uncertain conditions.

In automobile bearings the use of such high values for the "apparent" factor of safety (10 to 20), with a "real" factor of safety of 5, is advisable as a guarantee that unexpected load variations or accidental shocks will not cause strains beyond the elastic resilience of the materials.

The rings forming the races of a ball or roller bearing may be considered as circular beams, uniformly loaded between the points of support (the balls or rollers), with the balls or rollers at their points of contact with the surface of the race producing a pinning force ("wave" action) similar in effect to that produced by locomotive wheels passing or rolling on the steel rails.

In any radial or annular ball or roller bearing of either two point or four point of contact type the outer race has the same number of points of support or application of the load as the inner race, but the lineal distance or span between such points is greater since the lengths of the races (circumferences) are proportional to the diameters of the circles formed by them.

In a ball bearing the pinning stress upon the races, which is produced by the rolling of the balls under load, is much less marked on the inner race than on the outer, because the points of application of the load are closer together and it has practically continuous support on the shaft, especially if the shaft and bore of the bearing provide a tight drive fit. The outer

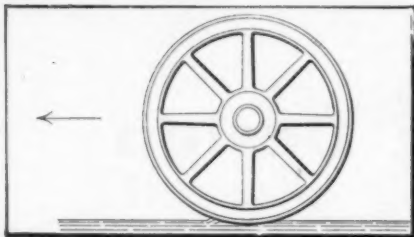


Fig. 1—Showing formation of asphalt "wave" in front of wheel of heavily loaded wagon in warm weather

contact with the balls, therefore, its thickness or cross-sectional area of metal should be greater than that of the inner race.

Neglecting the consideration of the continuous support which its contact with the shaft provides for the inner race, the relative thicknesses of metal in the outer and inner races of any annular ball bearing should vary in some ratio of proportion to the lengths of the chords  $s$  and  $S$  the lengths of spans between the points of contact between balls and races (Fig. 2).

The angle formed by any two fixed race surfaces in a 3-point or 4-point of contact type of ball bearing has a marked effect upon the frictional resistance to the rotation of the balls.

This frictional resistance to rotation varies as the  $\cos 1-2\phi$ , as illustrated in Fig. 3, and is due to the wedging effect of the ball in the angle of the race.

In amount it is equal to the effect of the load or pressure on the ball applied to a wedge having a thickness at the butt equal to  $\sin 1-2\phi \times \text{diam. of ball}$ , and a length equal to  $\secant 1-2\phi \times \text{radius of ball}$ .

Fig. 4 represents a wedge driven into the end of a log.

Let  $t$  = thickness of wedge =  $2 \sin 1-2\phi$  (Fig. 5)

$l$  = length of wedge =  $\cos 1-2\phi$

$W$  = resistance to splitting offered by the wood

$P$  = pressure or force required against the butt of the wedge to cause it to penetrate the log against the resistance  $W$ .

Neglecting the item of friction on the surface of the wedge,

$$\text{Then } P = \frac{W (2 \sin 1-2\phi)}{\cos 1-2\phi} = \frac{W \times t}{\cos 1-2\phi}$$

$$W = \frac{P (\cos 1-2\phi)}{2 \sin 1-2\phi}$$

If  $P$  and  $t$  are constant, then  $W$  varies directly as the length  $l$ ; therefore,  $W$  varies directly as the  $\cos 1-2\phi$ .

If a steel ball be placed between two hard steel surfaces or jaws, as in Fig. 6, then the wedging effect of any pressure  $P$  on the ball will be equal to that of a wedge formed by the tangents to the surface of the ball at its points of contact with the surfaces of the steel jaws, the length of which is therefore equal to  $\cos 1-2\phi$ .

If a ball bearing were constructed with this same angle  $\phi$  between the surfaces of the races in contact with the balls in a 4-point contact type, as illustrated in Fig. 7, then the frictional resistance to rotation of the ball, due to the binding or wedging of the ball between the race surfaces, will consume a very large proportion of the power applied.

Let  $f$  = frictional resistance to rotation of ball.

$P$  = pressure or load on the ball.

$r$  = radius of bearing = power arm of turning effort.

$\phi$  = angle between race surfaces.

Then:

$$f : P :: \cos 1-2\phi : 1 \dots (\text{When } r = 1)$$

Fig. 8 illustrates another extreme of this condition of contact between the ball and race surfaces. It is evident that when the angle  $\phi$ , included between the two race surfaces, is very obtuse, the resistance which is offered to the rotation of the ball by the wedging or binding effect of this angle becomes an almost negligible quantity.

Much time and space have been wasted in the discussion of

race, on the other hand, considered as a circular beam, has only its form and cross-sectional area of metal to resist the stresses and the increased tendency toward distortion which results from the greater intervals or spans between the points of contact

the theory that the pressure of the inner race, under load, tends to push the balls apart at the point of greatest intensity of load, and that there is a corresponding tendency of the balls to crowd together with considerable force in other portions of the circle.

One's imagination might create such a condition in a bearing in which the balls were of unequal sizes, under a moderate load which produced just sufficient elastic deformation of the larger balls to permit slight, but positive, contact between the race surfaces and the balls of smaller diameters without straining the large balls beyond the safe limits of proportionality. Under conditions of heavy load upon such a bearing the excessive strains to which the large balls would be subjected would probably damage or destroy both balls and races so rapidly and effectively that it would be difficult to trace the real cause to its source.

In any ball bearing in which the balls conform to the accepted standards of uniformity the balls are held in their relative positions by the full force of the reactions which occur under their load deformations, and the effects of slight inequalities of sphericity, or diameters, are compensated for in the variations of intensity of the load deformations.

During a series of tests of annular ball bearings of the 4-point contact type, 50 mm. bore, 110 mm. diameter and 27 mm. width, mounted on the

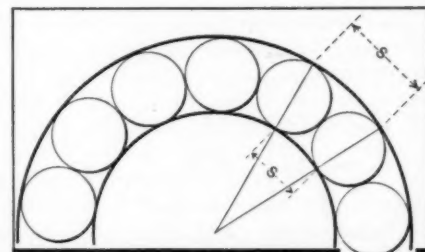


Fig. 2—Diagram illustrating proportional thickness of metal in outer and inner races of an annular ball bearing

same shaft with two No. 310 Hess-Bright bearings (of same dimensions), the deformation which occurs in the outer race under the effect of heavy loading was strikingly illustrated.

The bearings under test were provided with small separating rollers of about 5 mm. diameter between each pair of balls. The small axes, or trunnions, of these separating rollers, about 1 mm. or less in diameter, projected through small holes in a soft sheet-metal caging or retainer which was stamped out of ordinary No. 26 gauge sheet metal.

At one stage of the tests (after the bearings had been operated for several days at various speeds and loads) the machine was stopped for inspection, and the separating rollers, with the sheet-metal caging or retainer were removed from one of the bearings, without releasing the load of 4,000 pounds (2,000 pounds per bearing) which was being supported by the bearings.

The pull of 2,000 pounds per bearing was downward on the outer race, so that it was naturally expected that the balls in the lower, or unloaded, half of the bearing would roll together at the bottom. Not the slightest movement took place, however, and it was found impossible to move the balls from their relative positions by force of hand, when pushing against them as hard as possible with a piece of hard-wood hammer handle, or even when "jabbing" the stick against the balls; yet the bearings could be easily rotated by hand under the full load.

This indicates that the outer race (considered as a circular beam) is not only distorted, or deflected, under the load, between the points of support (balls) in the upper or loaded half of the bearing, but also that the reactions which take place in the lower or unloaded half cause similar deflections (although of lesser magnitude) in that portion of the outer ring or race.

Fig. 9 is an exaggerated illustration of the character of distortion which it is thus assumed takes place in the outer ring under the stress of heavy loading.

Upon removing the load the balls in the lower portion of the bearing immediately rolled together at the bottom.

In a ball bearing whose elements conform to the accepted standards of precision, the balls are held in their exact relative positions by the force of load reactions. During the tests above-

mentioned bearings with these small separating rollers made of commercial Stubb's wire (unhardened), supported by their small "pin" axles of about 1 mm. diameter (which in turn carried, or rotated, the soft sheet-metal caging) were operated continuously for several days at a time, under various loads, and at speeds up to several thousand revolutions per minute, without showing appreciable evidence of wear on either their small "pin" axles, or their minute bearings in the ordinary sheet-metal retainer; while the centers of the rollers, at their paths of contact with the balls, were more or less deeply grooved (probably 1-4 mm. in depth) to conform to the curvature of the ball surface, in all cases where "tight" assembly between the balls and rollers was made along the circle passing through their centers. No grooves, however, were formed in the rollers in the cases where a slight clearance, or tolerance, was allowed for between the balls and rollers, to provide for the "swelling" or deformation of the balls under the load.

Had there been the slightest tendency for the balls to shift from their proper relative positions during any part of the revolution the delicate little "pin" axles of the rollers, or the soft sheet-metal cages, must inevitably have become damaged during the many millions of revolutions made by them.

Professor Stribeck, in his report on "Ball Bearings for Various Loads" to the German small arms and ammunition factories, says:

"Hertz carried out a number of experiments with glass in order to prove the agreement between practical results and his deductions. He showed by pressing a glass lens against a plane disc of similar glass that the diameters of the pressure surfaces increased as the cube roots of the pressures.

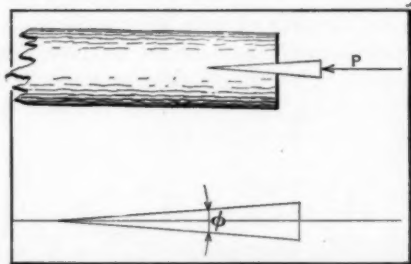
"To determine the influence of the elastic modulus, he pressed a steel lens against the plane surfaces of various materials, but encountering difficulties of observation he gave up the experimental proof.

"Later Auerbach, by also pressing lenses against plane discs, using three varieties of glass and one mountain crystal and measuring the compression areas, confirmed the relation of the compression areas, diameter, load, and the lens curvature radius

as previously investigated by Hertz.

"His experiments, however, with various curved bodies of the same material, gave no agreement of pressure at the elastic limit.

"According to Hertz, passing the elastic limit in glass or other similar brittle substances



Figs. 4 and 5—Illustrating effect of pressure on a ball by means of a wedge driven into the end of a log

first resulted in a circular crack closely following the edge of the pressure surface. The pressures causing these cracks were found to be dependent upon the curvature of the lenses, and invariably proportional to the ball surface diameters.

"So far there has been no basis from which to determine whether with metals, also, the pressure at the elastic limit is dependent upon the surface curvature; yet this problem is important enough in general, and more especially so for the designer of ball bearings. Its solution will define the relation of the permissible load to the curvature of the bodies in contact."

Prof. J. B. Johnson in his "Materials of Construction," says:

"The elastic limit of steel in compression marks the beginning of lateral flow of the metal, and the stress under which this action begins depends upon the freedom with which the metal can flow laterally."

Thus, in Fig. 10, we have a column compressed over its full cross-section, with freedom to flow laterally in every direction. This is the usual condition under which the elastic limit is found.

In Fig. 11 the specimen is compressed uniformly over a portion only of its surface, and, when the elastic limit is exceeded the metal finds escape by flowing laterally against the resistance of a ring of unstressed metal. This is a condition of restricted flow, and evidently the elastic limit is now much higher than before.

In Fig. 12 only the metal toward the center of the compressed surface is constrained to flow under direct stress, but in attempting to move laterally it is held by a ring of metal which is confined and compressed vertically, though inside its elastic limit.

To find escape the metal at the center must force its way against a much wider ring of metal than in the second case, and hence the elastic limit is now much higher than when pressed by a flat disc.

The elastic limit in compression, therefore, is a meaningless expression unless the conditions of lateral flow are also stated.

Experiments made by Prof. J. B. Johnson (see Trans., Am. Soc. C. E., Vol. XXXII, p. 270, 1894) to determine the areas of contact between a locomotive steel driving wheel (also cast-iron car wheels) and the cylindrical top surface of a steel rail (radius 14") developed the following:

1. That the area of contact increases directly with the load.
2. That the mean intensity of pressure is constant for all loads.
3. That, in these experiments, this mean intensity of compressive stress for all loads (5,000 to 50,000 lbs.) was about 82,000 lb. per square inch.
4. Since the maximum deformation (at the centers of these areas of contact) is twice the average deformation (assuming the volumetric deformation to be that of a segment of a paraboloid of revolution), then the maximum compressive stress intensity for all loads is about 164,000 lbs. per square inch.
5. Since no measurable permanent set was produced by any of these loads, on either wheels or rails, it follows that the "apparent elastic limits" of the materials had not been reached for this condition of contact, although the ordinary elastic limit of the rail material, for a free flow, was about 50,000 lbs. per square inch.

The experiments made by Professors Hertz, Stribeck and Auerbach tend to substantiate the deductions from Professor Johnson's tests, viz.:

That "Elastic Limit in Compression" is a meaningless expression unless the conditions of lateral flow are also stated.

That the compressive-stress intensity at the centers of the areas of contact between convex surfaces and discs is double the average, and in the case of the contact of the cylindrical rim of the locomotive drive-wheel with the curved surface of the rail, the maximum compressive-stress intensity was more than three times as great as the elastic limit of the rail steel under "free flow" without causing measurable permanent set.

The relative positions of the locomotive drive-wheels and the rail, at the point of contact, were those of two cylinders (of 44" and 28" diameter respectively) with their axes at right angle to each other—a condition under which the area of contact resulting from any pressure, or load, would be approximately the same as for a sphere of the smaller diameter pressed against a flat disc.

It is evident that the ratio of maximum intensity of compressive-stress to the elastic limit of the steel under free flow, without causing permanent set, will be even greater for the case of two balls pressed against each other than for the case of two cylinders with their axes at right angles.

Since the "apparent elastic limit" in compression between ball and ball is very much greater than for the

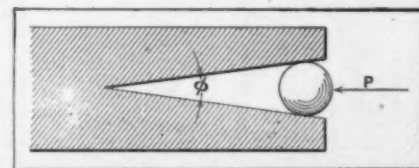


Fig. 6—Illustrating wedging effect of pressure on a ball wedged between steel jaws

same material under conditions of free flow, and between ball and disc is more than three times as great, it is evident that in designing ball bearings that form of race surface which permits the fulfillment of all other conditions with the least area of contact between ball and race under pressure should be selected. For very light loads and high speeds the sliding friction resulting from the larger area of contact between ball and curvilinear race (2-3 d. rad.) is greater than the resistance offered by convex or flat surfaces, but, as the loads increase, the resistance offered by the greater depth of elastic deformation with smaller area of contact, in the case of ball and convex surface, rises so rapidly as to outweigh all other considerations.

Theoretic considerations, therefore, would indicate that the three forms of races illustrated in Fig. 13 present different characteristics by reason of which each presents points of superiority for certain classes of service in the wide range of practical requirements, provided that sufficient thickness of metal is provided in the race so that the elastic deformation is localized at the surface, and the pinning effect of the rolling load is not compounded with bending or tension, deformations resulting from an insufficient area of metal in the race.

Thus convex race surfaces shown in "a" of Fig. 13 (although impractical from the commercial standpoint of manufacturing cost) are best adapted for very light loads where high speed is the essential consideration; "b" is best adapted for moderate loads and high speeds, and a wide range of practical conditions where alternations of moderate thrust loads are involved, or rigidity of lateral position is essential, and "c" is best adapted for very heavy loads at moderate or slow speeds.

The small actual area of contact between convex surfaces under pressure (even with the high coefficient of elastic deformation thus provided) makes this form impractical because of the relatively large sizes that would be required for even moderately heavy loads. Its limited field of usefulness, together with the mechanical difficulties and relatively high cost of manufacture, place this design outside the realm of practical consideration.

The designs illustrated in "b" and "c" of Fig. 13 readily adapt themselves to the requirements of almost any set of conditions in the entire range of practical application of ball bearings. Type "c" has the advantage of providing larger load capacity within given dimensions, while type "b" has the advantage of special adaptability to combinations of thrust and radial loads, without appreciable lateral motion or "end-play."

There is a wide neutral zone of usefulness in which either of these two types may serve with equal efficiency, and in which, therefore, the selection of one or the other type is merely a matter of practical considerations, such as limiting dimensions for necessary capacity, the cost, and personal preference.

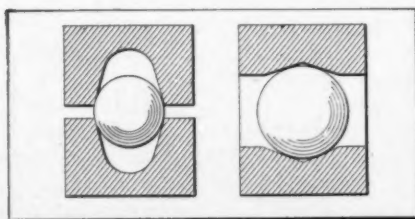
### Distribution of Load Stresses

In any ball bearing that is under load only the balls in one-half of the circle can be (theoretically) engaged at any one time, therefore, the entire load is assumed to be sustained by one-half of the total number of balls in the bearing.

Assuming that the total load "W" that is sustained by the bearing is so distributed along the projection of its horizontal diameter that the force of downward pressure at all points of contact between the outer race and the balls (as A, B, C, etc.) is of equal magnitude, and that this downward force at each

point of application (as B, C, D, etc.) is the resultant of two forces which act upon the same points in radial and tangential directions with reference to the circumference of the ball race.

Then, in Fig. 14, we may lay off the



Figs. 7 and 8—Illustrating frictional resistance to ball rotation in two types of four-point bearings

line P, on any scale, to represent the vertical pressure at B, and complete the parallelogram of forces.

Let b be the continuation of the line P to the horizontal diameter and r the radius corresponding to the direction of force P'.

Then, under the rule of similar triangles:

$$P : r :: P' : b$$

but,  $b = \cos \phi$ , in terms of r, therefore if  $r = 1$

then,

$$P' = P \cos \phi$$

1

When the ball is directly under the load, as at A,

Then angle  $\phi = 0^\circ$ , therefore  $\cos \phi = 1$

and,

$P' = P =$  maximum pressure on a single ball.

The total load on the bearing

$$W = P (1 + 2 \cos \phi + 2 \cos 2\phi + 2 \cos 3\phi \dots + 2 \cos n\phi) \quad 2$$

The sum of the cosines of the angles of all the balls in one-half of the bearing will therefore equal the load-factor K, which multiplied by the safe load capacity of one ball gives the maximum safe load which may be placed upon the bearing.

Thus, let K = load factor taken from Table I.

W = total load on the bearing.

P = maximum safe capacity of single ball from

Table II.

W

Then  $\frac{W}{K} =$  the greatest load which comes upon a single ball.

K

And  $P \times K =$  total safe load capacity of the bearing.

The assumption of load distribution (illustrated in Fig. 14) to produce equal vertical pressures or reactions at various points of unequal horizontal spacing does not accord with accepted mathematical theories for definite distribution of either uniform or concentrated load conditions, such as would obtain if the ring which forms the race of a ball bearing were considered as a continuous arch, a circular beam, a hoop, or other structural form for which definite and more or less satisfactory mathematical formulæ have been devised.

Fig. 9, and its accompanying description of practical tests, illustrates the nature of the forces and reactions which are transmitted around the entire circumference of the race from a static load acting in one direction only.

The reactions which result from the variable pinning force of the balls rolling under variable loads and speeds defy definite analysis, and destroy the accuracy of all calculations which are based upon other conditions alone, so that any assumption is

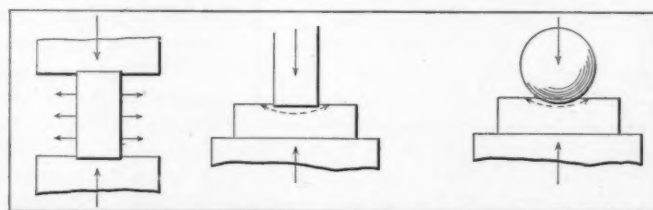


Fig. 10—Unrestricted lateral flow.

Fig. 11—Restricted lateral flow

Fig. 12—Confined flow

justified which indicates a distribution of load more nearly in accord with the practical conditions indicated.

The writer's experiments with ball bearings indicate that the stresses and reactions which occur in the loaded half of a ball race transmit stresses and reactions of much greater magnitude

TABLE I.—VALUES OF LOAD FACTOR K:

Number of Balls.	K.	Number of Balls.	K.	Number of Balls.	K.
6	2.00000	13	4.17942	20	6.31374
7	2.30411	14	4.49393	21	6.63460
8	2.61312	15	4.80973	22	6.95510
9	2.92371	16	5.12584	23	7.27547
10	3.23601	17	5.44240	24	7.59575
11	3.54947	18	5.75875	25	7.91582
12	3.86371	19	5.99273	26	8.23568

around the entire ring than is ordinarily assumed to be the case. The assumption of load distribution here given is therefore based merely upon its mathematical simplicity, and the fact that it indicates larger values for the radial pressures at the low points in the semicircle (such as C and D in Fig. 14) than would be indicated by assuming either a uniform or a concentrated load upon a beam having a length equal to the diameter of the enclosing circle, and calculating the pressures or reactions at intermediate points in the usual way.

#### Diameter of Enclosing Circle

To determine the diameter of the enclosing circle of any ring of balls:

Let  $n$  = number of balls.

$d$  = diameter of ball in inches.

$D'$  = diameter of enclosing circle in inches.

$\phi = 360 \text{ degrees} \div n$ .

$D''$  = diameter of circle through ball centers.

$x$  = clearance between balls, or space occupied by any form of separating device used.

$$\text{Then } D' = d + \frac{d + x}{\sin \frac{1}{2} \phi} \quad 3$$

In the full type of bearing take  $x = 0.005$  inch or more.

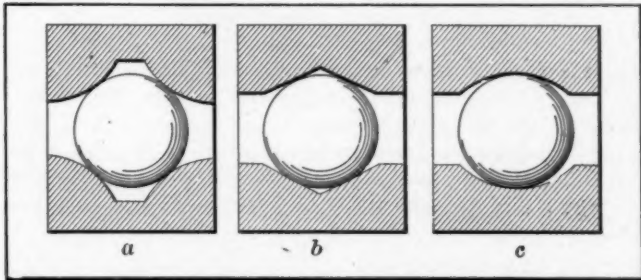


Fig. 13—Three forms of ball races: (a) Adapted for light loads and high speed; (b) moderate loads and high speed; (c) heavy loads and slow speed

If separating devices are used, then  $x$  = diameter, or thickness of one separating unit.

$$\text{Also } D' = K(d + x) + d \quad 4$$

In which the factor  $K$  is taken from Table I.

$$D'' = D' - d$$

$$x = 2 \sin \frac{1}{2} \phi \left( \frac{D' - d}{2} \right) - d$$

$$x = \sin \frac{1}{2} \phi (D' - d) - d \quad 5$$

$$\text{Or } x = \sin \frac{1}{2} \phi (D'' - d) \quad 6$$

#### Thickness of Metal in Races

The various stresses to which the outer race of a ball bearing is subjected by the pressure moment of the load are:

1st. Direct compression (elastic deformation) at the points of contact with the balls.

2nd. The pinning force which results from the rolling motion which occurs simultaneously with the application of compressive stresses of varying intensity, since each ball as it passes around the semicircle which represents the loaded half is subjected to a variation of direct pressure which (theoretically) varies from  $W$  to 0 and back to 0.

$W$

$K$

3rd. A tension stress which results from the tendency of the outer ring or race to deform from a true circle under the unequal application and distribution of the load.

The first and second stresses are greatest at the point of greatest intensity of direct load, which is generally at or near the bottom of the bearing, while the third, or tensile, stress is probably greatest on the opposite side of the ring (Fig. 15).

Let  $d$  = diameter of ball in inches.

$\phi$  = radial angle between centers of adjacent balls.

$P$  = safe load capacity of one ball.

$E$  = elastic limit of steel used.

$S$  = safe working stress for the steel under the conditions of load and speed to be provided for.

$M$  = minimum thickness of metal in the outer race.

$m$  = minimum thickness of metal in the inner race.

$c$  = effective width of race.

$D$  = outside diameter of the bearing.

$D'$  = diameter ball path in outer race (or the enclosing circle).

$D''$  = diameter of circle through ball centers.

$D'''$  = diameter ball path in inner race.

$R, R', R''$  and  $R'''$  = radii corresponding to  $D, D', D''$  and  $D'''$ .

$W$  = total load capacity of bearing.

For high-grade alloy steels, such as are generally used for ball bearings, the elastic limit  $E$  is generally about 100,000 pounds per square inch; therefore, with this class of material the safe working stress  $S$  may be taken as follows:

$$S = \frac{100,000}{5}$$

= 20,000 for bearings under quiescent load, or for very low speed and steady load.

$$S = \frac{100,000}{10}$$

= 10,000 for variable live loads and speeds.

$$S = \frac{100,000}{20}$$

= 5,000 for bearings subject to shocks.

The minimum thickness of metal ( $M$  and  $m$ ) to be provided in the races of a ball bearing which is designed for stresses at the maximum safe load capacity of the balls will be:

$$M = \frac{6 P R' 2 \sin \frac{1}{2} \phi}{10 S c}$$

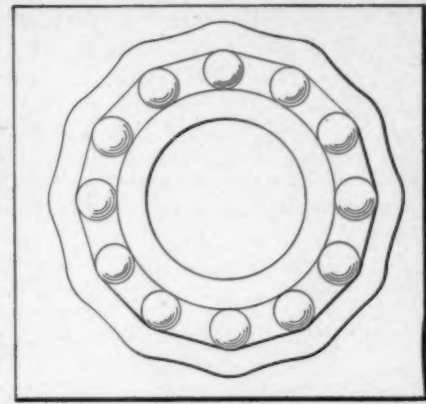


Fig. 9—Illustrating distortion which takes place in outer ring under heavy loading

TABLE II.—PHYSICAL PROPERTIES OF HIGH-GRADE STEEL BALLS

DIAMETER OF ALL D.			CRUSHING STRENGTH BETWEEN			SAFE LOAD PER BALL FOR		
Inches	Inches	Mm.	d <sup>2</sup> Inches	Ball on Ball 60000d	Flat Plates 80000d <sup>2</sup>	3 d Grooves 100000 d <sup>2</sup>	Lbs.	Lbs.
1/16	.25	6.25	.0625	3,750	5,000	6,250	125	83.3
1/8	.3125	7.937	.0795	4,746	6,328	7,910	158	105.5
3/16	.3750	9.525	.1406	8,436	18,248	14,063	280	194.2
1/4	.5000	12.700	.2500	15,000	20,000	25,000	500	333.3
5/16	.6250	15.875	.3906	23,446	31,248	39,062	781	520.8
3/8	.7500	19.050	.5625	33,750	45,000	56,250	1125	750.0
7/16	.8750	22.225	.7656	45,936	61,248	76,562	1531	1020.6
1/2	1.0000	25.400	1.0000	60,000	80,000	100,000	2000	1333.3
5/8	1.125	28.575	1.2656	75,936	101,248	126,560	2531	1687.5
3/4	1.250	31.750	1.5625	93,750	125,000	156,250	3125	2083.3
7/8	1.375	34.925	1.8789	113,750	150,000	187,500	3906	2591.7
1	1.500	38.100	2.2500	135,000	180,000	225,000	4500	3000.0

$$M = \sqrt{\frac{6 P D' \sin \frac{1}{2} \phi}{10 S c}} \quad 7$$

$$m = \sqrt{\frac{6 P D'' \sin \frac{1}{2} \phi}{10 S c}} \quad 8$$

In terms of  $d$ , for the maximum safe load capacity, the minimum thicknesses of  $M$  and  $m$  may be determined by the following:

$$M = d \sqrt{\frac{12,000 D' \sin \frac{1}{2} \phi}{10 S c}} \text{ for two-point type.} \quad 9$$

$$M = d \sqrt{\frac{8,000 D' \sin \frac{1}{2} \phi}{10 S c}} \text{ for four-point type.} \quad 10$$

Thus in bearings of equal dimensions and with the same number of balls, the four-point type of bearing requires less thickness of metal in the races, because the safe working load capacity for balls for that form of contact is only about two-thirds as large as for two-point grooved races, and the total load capacity of the four-point type of bearing must be further reduced by multiplying the apparent load capacity by the cosine of the angle of inclination of the race surface. ( $\cos 0^\circ = 1$ ,  $\cos 90^\circ = 0$ ,  $\cos 45^\circ = .7$ ).

To determine the minimum thickness of metal ( $M$  and  $m$ ) for any desired working load  $W$ , the following formulæ may be used:

$$M = \sqrt{\frac{3 W D' \sin \frac{1}{2} \phi}{5 S c K}} \quad 11$$

Or,  $M = \sin \frac{1}{2} \phi \sqrt{\frac{3 W D'}{5 S c}} \quad 12$

$$m = \sqrt{\frac{3 W D'' \sin \frac{1}{2} \phi}{5 S c K}} \quad 13$$

Increasing the number of balls by using balls of smaller diameter permits the use of heavier races within the limiting outer diameter of the bearing and decreases the maximum load per ball.

In the formula  $W/K$ , which determines the maximum load which comes upon a single ball, the denominator  $K$  increases as the number of balls is increased (being approximately one-third of  $n$ ); hence inversely as the diameter of the balls used; while the numerator  $W$ , or total safe load capacity of the bearing ( $= P \times K$ ), increases as a variable function of the square of the diameter of balls used; therefore, the largest size of balls permitted by the limiting bore and diameter of housing will give the largest rated load capacity for the bearing.

#### Effect of Speed Upon Load Capacity

To increase the size of balls to be used in a ball bearing, however, also involves an increase in the thickness of metal required for the races.

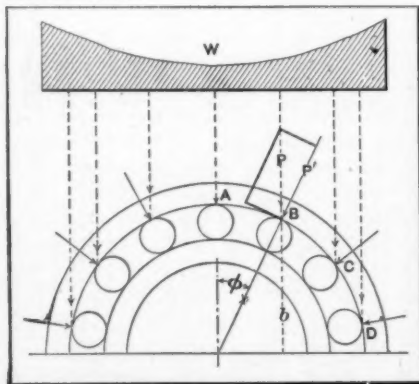


Fig. 14—Diagram illustrating the assumption of load distribution in a ball bearing

In his résumé of Professor Striebeck's report on a series of tests on ball bearings Henry Hess (Trans. A. S. M. E., Vol. 28) makes the following assertion:

"Speed of rotation, in so far as it is uniform, does not affect the carrying capacity. (This applies to radial bearings, but not to thrust bearings of

the collar type; in these the carrying capacity decreases with increase of speed.)"

This assertion is contrary to fundamental principles of mechanics and is, therefore, absurd. It is true that in a series of practical tests of ball bearings, involving moderate variations of load and speed, the effect of gradual speed variation is negligible, because with the high factor of safety necessarily used in proportioning ball bearings the entire range of such moderate speed variation merely affects the magnitude of the safety factor and the stresses are at all stages well within the elastic resilience of the materials.

The writer's limited range of experiments indicates that the load capacity of a radial four-point contact type of ball bearing decreases in a somewhat more rapid ratio than the cube-roots of the speeds and at a less rapid rate than the square-roots of the speeds; therefore, in the absence of data admitting of a greater degree of refinement, the following empirical rule is suggested.

For ball bearings of the four-point contact type, under radial loads, the safe load capacity decreases inversely as the mean of the square root and cube root of increased speeds.

Adopting 200 revolutions per minute as the normal speed below which corrections of safe load capacity for speed variation are negligible,

$$\frac{\sqrt[2]{\frac{200}{S}} + \sqrt[3]{\frac{200}{S'}}}{2} = 10 \text{ (approximately),}$$

then the corrected safe load capacity of a radial ball bearing, for any speed above 200 revolutions per minute, may be determined by the following formula:

Let  $L$  = maximum rated load capacity given in catalogues.

$S'$  = speed in revolutions per minute under consideration.

$L'$  = safe load capacity at the desired speed  $S'$ .

Then

$$L' = \frac{10 L}{\sqrt[2]{S'} + \sqrt[3]{S'}} \quad 14$$

For convenience, a number of values of the denominator in the above formula are given in table III.

#### Designing Roller Bearings

For two cylinders in contact with their axes parallel the form of contact area is theoretically a line, but practically a parallelogram of infinitesimal width and of a length equal to that of the roller.

For ball and disc the form of contact area is practically a circle of infinitesimal diameter.

Under the effect of load deformations the roller contact area increases in one direction only (width), while the ball contact area increases radially in all directions, or, as  $\pi r^2$ .

Let  $\delta$  = deformation, or reduction of radius.

$\theta$  = angle, of which  $r - \delta = \cosine$ .

$B_a$  = area of ball contact surface.

$R_a$  = area of roller contact surface.

$B_p$  = safe load capacity of ball.

$R_p$  = safe load capacity of roller.

TABLE 3.—SPEED FACTORS.

R.P.M.	$\frac{\sqrt[2]{S'} + \sqrt[3]{S'}}{2}$	R.P.M.	$\frac{\sqrt[2]{S'} + \sqrt[3]{S'}}{2}$	R.P.M.	$\frac{\sqrt[2]{S'} + \sqrt[3]{S'}}{2}$
200	10	900	19.82	4000	39.57
300	12.00	1000	20.81	5000	43.90
400	13.68	1500	25.10	6000	47.85
500	15.15	2000	29.43	7000	51.40
600	16.46	2500	32.80	8000	54.7
700	17.67	3000	34.6	9000	57.85
800	18.78	3500	37.2	10000	60.78

Then for ball and roller of equal dimensions (i. e., roller having diameter and length = diameter of ball).

$$B_a = \pi (r \sin \frac{1}{2} \theta)^2$$

$$R_a = 2 r \sin \frac{1}{2} \theta \times d$$

$$R_a : B_a :: 2 r \sin \frac{1}{2} \theta d : \pi (r \sin \frac{1}{2} \theta)^2$$

$$R_a = B_a \frac{2 r \sin \frac{1}{2} \theta d}{\pi (r \sin \frac{1}{2} \theta)^2}$$

$$R_a = B_a \frac{1.273}{\sin \frac{1}{2} \theta}$$

Since the loads are directly proportional to the areas of contact, and for ball and disc, the contact area (according to Professor Hertz) increases as the cube root of the pressure, then:

$$R_p : B_p :: \frac{1.273 d}{\sin \frac{1}{2} \theta} : d$$

$$R_p = B_p \sqrt[3]{\frac{1.273}{\sin \frac{1}{2} \theta}}$$

For  $\delta = .00001 r$ ,  $.99999 = \cos 0^\circ 10' = \cos \frac{1}{2} \theta$ .  $\sin \frac{1}{2} \theta = .00291$

$$R_p = B_p \sqrt[3]{437} = \text{Approx. } 7.59 B_p$$

For  $\delta = .00002 r$ ,  $\sin \frac{1}{2} \theta \sin 0^\circ 20' = .00582$

$$R_p = B_p \sqrt[3]{218} = \text{Approx. } 6.02 B_p$$

For  $\delta = .00015 r$ ,  $\sin \frac{1}{2} \theta \sin 1^\circ 00' = .01745$

$$R_p = B_p \sqrt[3]{73} = \text{Approx. } 4.17 B_p$$

For  $\delta = .001 r$ ,  $\sin \frac{1}{2} \theta \sin 2^\circ 30' (\text{Approx.}) = .04362$

$$R_p = B_p \sqrt[3]{29} = \text{Approx. } 3.07 B_p$$

This indicates that for extreme loads, or for stresses beyond the limits of proportionality, the advantage of greater load capacity of a roller, as compared with that of a ball of the same diameter, diminishes very rapidly.

In designing a roller bearing the safe load capacity of one roller may be determined as follows:

Let  $d$  = diameter of both ball and roller, in ins.

$L$  = length of roller in ins.

$P$  = safe load capacity of ball from Table II.

$P'$  = safe load capacity of roller of diam. and length =  $d$ .

Then

$$P' = \frac{7.5 P L}{d} \quad (\text{for light loads}) \quad 15$$

or

$$P' = \frac{6 P L}{d} \quad (\text{for heavy loads}) \quad 16$$

The general formulæ, Nos. 1 to 6, may be applied to determine the load distribution, diameter of enclosing circle, etc., of the roller bearing, while formulæ 7 and 8, for determining the thickness of metal required in ball races, may be applied to roller bearings by substituting the value of  $P'$  (as determined by formula 15 or 16) for  $P$  in these formulæ.

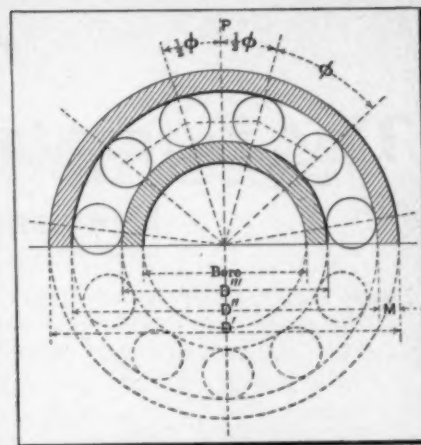


Fig. 15—Diagram illustrating direct compression, pinning force and tension stress in a ball bearing

## Long- vs. Short-Stroke Motors

By Justus B. Entz, Member S. A. E.

*Paper read at the summer meeting of the Society of Automobile Engineers, at Dayton, O., June 15-17, discussing the relative merits of the two types of motors under varying conditions.*

IN considering the relation of bore to stroke in an automobile gas engine we may assume that at 1000 revolutions per minute the power obtained will be proportional to the piston displacement, and that a  $4\frac{1}{2} \times 4\frac{1}{2}$  and a  $4 \times 5\frac{3}{4}$  will give equal power. If each engine is designed for the same percentage of compression space, and has valves proportional in size to the bore, we find that the shorter stroke engine has a total pressure on its piston head of  $5\frac{3}{4} \div 4\frac{1}{2}$ , or 26 per cent. more than the longer stroke engine, and that as the crankshaft and connecting-rod bearings turn but once in their boxes per revolution, whether the stroke be long or short, the loss in them is increased.

The side pressure of the piston on the cylinder walls is also greater in the same proportion, but as the piston speed is correspondingly less, this loss will be about the same in each. But the result is a higher mechanical efficiency for the long stroke. The piston in the long stroke is lighter, being less in head and wall, but the speed being higher the balance of the two engines at the same revolutions per minute will probably not differ much.

The wall area of the compression space is less in the long stroke, and its thermal efficiency is higher therefore, as well as its

mechanical efficiency. The torque of the long stroke is higher at low speeds due to its higher thermal efficiency, which is lowest at low speeds.

At high speeds, however, as the same volume must be drawn through smaller valves, the long stroke will have a less volumetric efficiency, if the valves are small enough to be the limiting factor, and therefore will lose power at high speeds. But in practice even in engines of 40 per cent. longer stroke than bore, the carbureter generally determines the volumetric efficiency rather than the valves.

It is, of course, a question as to how much the stroke can be increased as compared with the bore and give better results, but I believe that engines with a stroke relation to bore of from 1.4 to 1.5 are lighter, more efficient, and more flexible than shorter stroke engines.

**GUADELOUPE OFFERS A MARKET.**—Guadeloupe roads are very bad as a rule, and inasmuch as it is claimed by some of the Guadeloupeans that American-made motor cars stand up better under rough road conditions than automobiles which are manufactured in France, there is said to be a fair show for the purchase of American-made machines in the not very remote future. But as Guadeloupe maintains no electric plant, automobiles propelled by electricity are out of the question. At the same time, there seems to be little encouragement for gasoline machines, owing to the irregular supply of this commodity.

# The Society of Automobile Engineers

## Discussing Its International Scope

*Henry Hess, past president of the Society of Automobile Engineers, states some pertinent facts in relation to the formation and past doings of the organization. It is pointed out that the society is unique among organizations of a technical character, and that the membership is enthusiastic and exerts a wide influence by its team work. As Mr. Hess says: "The world is to-day but small. International intercourse is quick and easy." The work of the society is on an international basis.*

**A**MONG the principal engineering societies of the United States the "Society of Automobile Engineers" may fairly be held to have made itself a place, and a factor in a most important industry.

"Coming events cast their shadows before!"

The Society of Automobile Engineers is unique among technical societies in its title—it has no national prefix. Was the omission pure accident or was it prophetic of a destined international scope? It would almost seem the latter, in view of developments that are close at hand that have, in fact, already set in.

Starting with but a handful of enthusiastic members, each year saw the society double its membership with increasing ease as its work and meetings could in turn take wider ground and till that ground more thoroughly.

When the Mechanical Branch of the Licensed Automobile Manufacturers' Association ceased from being, the opportunity was promptly seized and overtures made and promptly accepted to take up this dropped work. This brought a further accession to the membership in just recognition of the fact that the society was in even better position to carry on the work without bias or interference from any commercial dictation, but working harmoniously with and for the best interests of the entire community interested in the use, making or design of the automobile.

With a view to giving every man and every firm in the community having an interest in the automobile from any point of view an opportunity to work to the mutual advantage of all and to so work within and through the society and to thus secure the cumulative benefit of such collective working, the requirements for admission to membership were liberalized for members, associates and juniors, and a new grade created to admit corporate bodies, termed affiliate membership and affiliate representation. The immediate response in membership increase proved the wisdom of these changes.

The society has taken up as one phase of its activity the standardization of parts and elements entering into automobile construction. It has been possible within the year for its members, working as special committees, to make recommendations that have resulted in great economies to all concerned. It is to be remembered that the society is wisely forbidden by its constitution to adopt any or recommend any standard. All such recommendations are personal to the members of committees. The society simply forms the vehicle for the centralizing of the knowledge and skill available in the profession and brings that to bear on a given problem; the society further provides the machinery for the distribution to the entire community of the results of this concentrated work.

There are now many committees at work dealing with many subjects. Some of these have found it easy to reach a result,

others have not. The underlying cause of this difference is simple: Whenever an element is supplied by but few manufacturers, but with many small variations gradually brought about in response to individual wishes and specifications, these manufacturers are only too anxious to secure the adoption of a relatively small range of variation; they will offer in inducement the sharing in the reduction in cost that will result from such simplification. In other words, all parties are alike anxious to get together and, as whatever benefits the one benefits the other, agreement is soon reached.

But let the distribution of benefits not be equal, then agreement will be much more difficult to reach. Each side will naturally want the most and wish to give the least. As a case in point, the matter of tolerances for securing interchangeability may be cited and it may be assumed that a total limit between the piece to fit and the piece to be fitted to of ten units has been agreed upon: The purchaser will find it cheapest to take the entire tolerance for himself; but so will the seller. As the seller makes his parts in by far the largest quantity (since he supplies many purchasers) he will have provided many and costly tools and limit gauges, which he cannot very well discard owing to the great financial loss involved. The matter becomes still more complicated when the same parts are made by a number of manufacturers, each having a different set of tolerances. Difficult as is a solution under these conditions, it can be brought about with time and patient working. This brings me to the "International Scope" referred to in the title.

The world is to-day but small. International intercourse is quick and easy. Import and export more and more are determined by final advantage, uninfluenced by frontiers and ocean barriers. That makes international standardization a necessity in many things. In some articles, standardization on national lines only is impossible. Assume our case as above, but with the additional fact that the article in question is made in one country and used in many others. Obviously the tolerances everywhere must be alike or the makers will be compelled to get up the same article to many different specifications. If the article be one made for stock, that means also the carrying of large stocks to different specifications. That means greater costs to manufacturer and consumer and all along the line. But more than that; the maker may have in stock a few thousand to French and German and British standards, and yet be unable to supply a rush demand from here, owing to a difference of standard. Were there but one international standard, that would act as a balance wheel and make the stock available to rush demand from any source. The situation becomes still more complicated when the same article is made by a number of manufacturers in a number of different countries. This is not a fancied difficulty, but a very real one, as a number of committees now dealing with this knotty problem can testify.

There is a remedy—the internationalization of engineering societies in certain phases of their activities. A movement is on foot for the further broadening of this tendency by bringing into existence a body that will join the efforts of all technical societies of all nations. Necessarily the work will be slow; must, in fact, be as slow as it is far-reaching. But as the work responds to a real and an increasing need it will gather momentum as it goes on.

The Society of Automobile Engineers has the honor of being the first to start this movement of international standardization by the appointment of a committee to deal with this phase

# Piston and Ring Oversize Standards

By James N. Heald, Member S. A. E.

*Paper read at the Summer Meeting of the Society of Automobile Engineers, Dayton, O., June 15-17, dealing with the advantages to be derived from the making and carrying in stock at factory and branch houses of standard oversize pistons and piston rings, so that in the regrinding of a scored cylinder close-fitting pistons and rings shall be ready at hand to complete the repair. Showing also the advantage to the repairman of having his shop equipped with a cylinder grinder, with the resulting benefit to the owner in the lessening of the time while his car is out of commission.*

FOR several years past the automobile and gas engine manufacturer has been so busily engaged in the development and perfecting of his machines and meeting the demand upon him for his product that he has, up to the present time, apparently given very little thought to the ways and means for reducing the cost of maintenance among his customers, as well as for preventing the delays so frequently met with in repairs that are required from time to time.

Thousands and thousands of people all over the country have taken up the automobile very enthusiastically, but there are many users of cars to-day who can ill afford them, due principally to the cost of up-keep; and after a time they cease to be purchasers of cars and equipment.

When the owner of a car finds that he has a scored cylinder or that the cylinders are badly worn he is face to face with a difficult problem to solve to his satisfaction. He can send to the manufacturer for a new cylinder, or he can have his old cylinder refinished and have new pistons and rings made up in some local machine shop; but either method has great disadvantages, both in regard to the time that his car is out of service and also with regard to the expense of the repairs to be made.

The matter of expense for various items in connection with the purchase and maintenance of an automobile is one that ought to receive more thought than apparently has been the case, for the general good of the automobile industry.

The purchaser of a car is held up in so many ways that it is discouraging to one who thinks seriously of buying a car; he may decide that he can afford to spend a certain sum, but when he comes to look into the question carefully, if he pays about that sum for the car, he often finds he has still a top to buy, a wind shield, a gas tank and other items of equipment, to say nothing of the things it is convenient to have in the garage (which also may have to be built) in the way of gasoline tank, oil cans and accessories. Then if any repairs are needed the cost is seemingly out of proportion to what it should be.

The point of interest in this is that there are so many expenses connected with the running of a car that it would seem a desirable thing for the manufacturer and the repair man to adopt any plan by which the time and expenses of making repairs to an engine are reduced to the lowest point. For that reason the writer feels that the adoption of the scheme to be outlined in this paper will be of great advantage in bringing about this desirable result.

As already stated, if the owner of a car who finds a cylinder badly scored concludes to send to the factory for a new cylinder he is usually called upon to wait a considerable length of time,

first, because such orders usually take their turn in being filled, which requires a certain amount of routine work in the factory in the way of entering orders and boxing and shipping out the same; and, secondly, the time required for the order to go to the factory and the goods to come to the purchaser seems considerable if the manufacturer is five hundred or a thousand miles away, especially if one is in a hurry to get his car in commission.

Furthermore, as many of the engines are now made with cylinders cast in pairs, or en bloc, an imperfection in one cylinder bore makes it necessary to throw away cylinders in which the other bores are in perfect condition, in fact, in better condition than any new cylinder which will come from the factory can be, because the old rings and pistons will not usually fit the new cylinder as well, and in any event the new cylinder is comparatively rough, and does not have the excellent bearing and contact with the rings which was the case with the holes in the cylinder discarded.

For that reason the ability to refinish a cylinder by the repair man is most desirable and means a considerable saving in time and money to the owner of a car.

The advantages of being able to refinish cylinders are not confined to those which have become scored up in running, but apply equally in the case of ordinary wear, which is found in all engines after running ten, twenty or thirty thousand miles.

In this case the refinishing of the cylinders and the substitution of new pistons and rings also means the putting of the engine in first-class condition at a minimum cost.

The refinishing of a cylinder bore can be most easily accomplished by the use of a cylinder grinder, such as have been used for a good many years by almost all the leading concerns in the manufacture of automobile and gas engines.

The process of grinding a cylinder to refinish it is extremely simple and much easier to handle than the reboring of cylinders in a boring mill or by any other method. It is a very easy matter to mount a cylinder on one of these grinding machines, as all that is necessary is to clamp it against an angle plate by the crank end of the cylinder, and any amount of stock can be

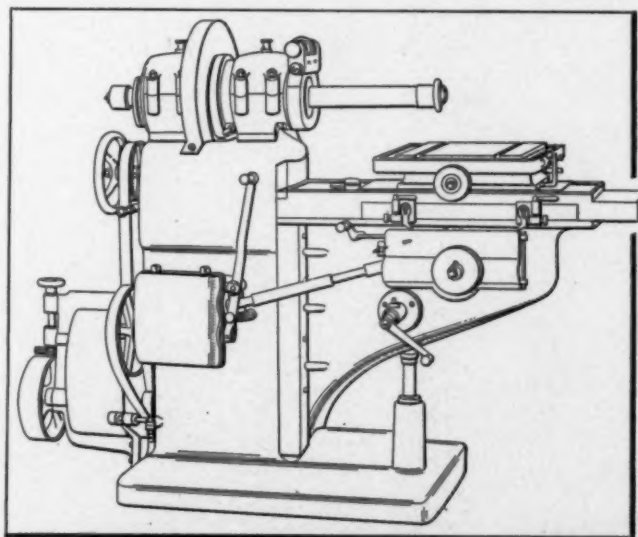


Fig. 1—Type of cylinder grinder which should form part of the equipment of the up-to-date repair shop

removed that is necessary to produce a perfect hole and remove the scoring or other imperfections that may exist.

The use of the grinding machine makes it possible to enlarge the hole any desired amount, whether it be ten-thousandths of an inch or twenty-thousandths or fifty-thousandths, and the grinding wheel finds no difficulty in removing the smooth and polished surface found in a cylinder after it has been run for a time, whereas on a boring mill a heavy cut must usually be taken in order that the tool can get underneath this polished surface, which means, of course, that the bore is then unduly enlarged, and, moreover, a comparatively rough surface is left by the boring tool, which is not the case with refinishing on a grinding machine.

A further advantage in using the grinding machine is that it will handle the cylinder without its being rotated; that is, a cylinder with four holes, for instance, can be bolted against the angle plate and ground out as easily as though it was a casting with a single hole (Fig. 3), while if this was put on a boring mill one of exceedingly large size would be necessary in order to have room to swing so large a casting.

Oftentimes cylinders are injured by having a corner of the cylinder and flange broken off, and very frequently cylinders are brought to the shop to be refinished in which these broken pieces have been brazed on again or joined by autogenous welding—in fact, there are now many plants about the country advertising and offering to do this work promptly and cheaply.

In this case the usual condition is that the bore of the cylinder, except at the very end near the crank-base, is in first-class condition, and no new pistons or rings are needed, but it is important to refinish the lower end of the hole where the brazing has been done, because this usually leaves a very rough and uneven surface, and oftentimes as much as one-eighth of an inch in spots must be removed to get down to the tube surface, so that the piston and rings can again enter the bore.

In such cases the grinding machine is of the greatest possible advantage, because the cylinder can be located in the machine and adjusted to such a position that the grinding wheel will remove only the surplus stock which the welding process has put on, and this can be made absolutely concentric with the balance of the hole, and the cylinder made as perfect and as satisfactory as though no accident had ever occurred.

This can be done at a fraction of the cost of a new cylinder; in fact, in general terms the cost to the customer does not exceed two or three dollars per hole, even where the entire cylinder is refinished to ten or fifteen-thousandths over the original size.

The general style and appearance of the cylinder grinding machine referred to is shown in Fig. 1. I also show a view of the machine with a cylinder mounted on a jig or angle plate, looking into the two holes which are just being finished by this process (Fig. 2).

The cut shows the grinding wheel just out of sight in the further head. The grinding head, carrying the wheel and spindle, is so constructed that the wheel is made to travel in a perfect circle, as well as rotate on its own axis, enabling it in the hands of the average operator to grind a perfectly round hole.

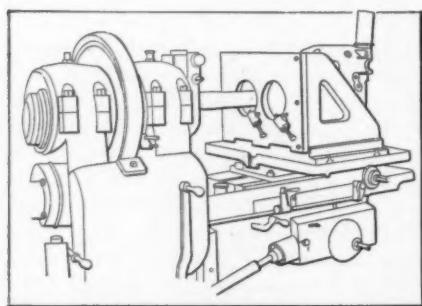


Fig. 2—Showing a cylinder grinder operating on a two-cylinder casting

This grinding head is made up of two eccentrics, one within the other; the relative adjustment of these eccentrics by the small knob, shown in the upper center of the picture, produces a larger or smaller throw to the eccentrics, and, therefore, makes the wheel travel in a larger or smaller circle, as de-

sired, for different diameters of hole, and also enables the operator to increase the throw to bring the wheel to the work and to make up for the wearing out of the wheel, which, of course, occurs very slowly in the regular operation of the machine.

**USING OLD CYLINDER OF ENLARGED BORE**—Occasionally in refishing a cylinder the scoring is so slight that an increase in diameter of perhaps five or six-thousandths of an inch only is necessary to produce a perfect hole. In such cases it is often considered a question whether a new piston is necessary, and frequently customers will decide to use the old pistons, together with a set of rings which are made enough larger to make up for the increased diameter of the bore. Just how much larger the bore can be made than the standard size and have the old piston prove satisfactory to the average user is a debatable question.

An examination of some of the forces acting in the cylinder may throw some light on this problem; in the accompanying diagram (Fig. 4) it will be noticed that when the piston is descending, due to the force of the explosion, it is pressed against the side of the cylinder, A, owing to the angularity of the connecting rod, B-C, in its downward travel, but when the crank has made a half revolution and the crank-pin is at the position D the resistance above the piston to its upward movement produces a pressure of the piston against the wall on the opposite side, as at E, and for that reason it is generally considered, as we understand it, that when there is too much clearance between the size of the piston and the cylinder bore the engine will become noisy on account of the alternate crowding of the piston against one side and then against the other side of the cylinder at every half revolution.

When there is much difference in size between piston and cylinder, of course the opportunity for leakage of gas through the cut in the ring is increased, and it becomes more difficult to maintain compression, especially when the rings are worn a little and the cylinder walls have worn a little, and the opening where the ring is split has become of some little width.

**WHEN NEW PISTONS ARE DESIRABLE**—It therefore seems desirable to make new pistons and rings to secure a quiet running engine when any considerable amount of stock is removed in the refinishing of the cylinder bores. From our experience we would set the limit of increase in size to five or six one-thousandths of an inch if old pistons are to be used.

**GETTING OVERSIZE PISTONS**—Now new pistons being decided upon, the question comes as to securing these. As we understand it, very few of the manufacturers of engines are prepared to furnish anything but the standard size pistons.

They are also not particularly anxious to furnish rough castings to the repair man to be used in making pistons of oversize dimensions, and even if they were, too much time is required to send to the factory to secure these.

For that reason the job shop is called upon to make a pattern for a piston and to fit up the castings. The cost of the pattern is a variable quantity; if a good many castings were wanted, of course, it would pay to make a reasonably good pattern, but if there are only one or two pistons wanted, as is usually the case, one feels like making a pattern as cheaply as possible.

Oftentimes a shop which has had considerable of this work will look over its patterns and find a pattern which comes somewhere near in dimensions to that which is wanted at the time, and with a little patching or fixing that pattern is made to serve. But the casting which is secured from this pattern may vary largely from the correct dimensions and weigh half a pound

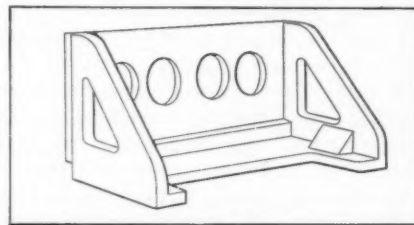


Fig. 3—Four-cylinder casting, can be handled as easily as a single hole

or a pound more than the other pistons in the engine, which cannot be conducive to smooth running when the engine is assembled with this odd-sized piston in the outfit.

**MACHINING**—After this casting is secured the question of machining the piston comes up, and while this is an easy thing to turn up to approximately the right size, the putting of the hole through for the piston pin at an exactly right angle to the center line of the piston is a problem which has too little consideration usually; and unless this is correctly done by the piston being mounted properly on the face plate of a lathe and with suitable jigs and tools the chances are ten to one, or perhaps more nearly one hundred to one, that the piston pin will not be at right angle, and the connecting rod will not stand in proper relation to the crank-pin when an effort is made by the repairman to assemble the engine. This means a noisy engine, one that quickly starts to pound and is hard to keep in smooth running adjustment despite all attempts to the contrary. Therefore it seems that the adoption and furnishing of certain oversize standards of pistons and rings would be an extremely desirable thing for the manufacturer of engines to arrange.

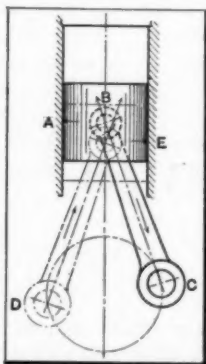


Fig. 4—Diagram showing how noise is produced by an ill-fitting piston

bore refinished to ten, twenty or fifty-thousandths over standard size, and the engine could at once be reassembled with the new piston and rings with the least possible delay and expense for this work.

The pistons could be made by the manufacturer to weigh exactly the same as the standard piston, and therefore a new piston assembled in an engine would run as smoothly with regard to weight of reciprocating parts as was the case when the engine was first assembled.

The item of expense is an important one, and the contrast between these two methods is notable; the jobshop man figures that the pattern for the piston will cost him \$2.50 to \$4, according to the amount of care taken in making it; the cost of turning up a piston and making a set of rings will amount to from \$10 to \$20, or even more, according to the shop doing the work, per piston and set of rings, and this covers work of really the roughest character only, as compared with what is produced by first-class manufacturing methods in a well-ordered manufacturing plant; on the other hand, the regrinding of a cylinder will cost on an average from \$2 to \$3, and a new piston and rings, if made up according to manufacturing methods, certainly could be sold for only a fraction of the cost necessary where a single piston and set of rings are made up in a job shop without proper facilities and to special order.

We understand that at the present time the Pierce-Arrow Motor Car Company (and possibly others of whom we have not learned) are furnishing these oversize standards in ten, twenty and thirty-thousandths of an inch large.

Also that the cost to the customer for a piston and set of rings with pin and screws complete for one cylinder is \$5.70 for the four-cylinder and \$6 for the six-cylinder style.

**AN UNUSUAL JOB OF CYLINDER GRINDING**—This indicates in the most plain and convincing manner the saving in cost that can be made by this method of handling such repairs; and the

gain in time is equally as great and acceptable to the car owner.

Another advantage that would accrue to the manufacturers of cars would be in being able to advertise the fact that they were in position to furnish repairs at a moderate cost, and their agents could make repairs if needed at the lowest possible cost.

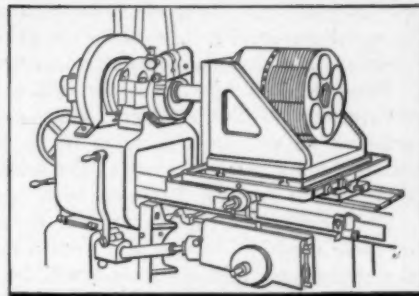


Fig. 5—Cylinder grinder operating on a six-cylinder block casting

This is an item that must be reckoned with in the future selling of cars by the commercial department of car manufacturers, as it is certainly one that is being given more and more consideration by the intelligent buyer, and it would seem that it would have much influence in determining which car he would purchase.

**WHAT THE OVERSIZE STANDARDS SHOULD BE**—The question of how many and just what the oversize standards should be is a matter that is open to discussion; it is our experience, however, in regrinding hundreds and hundreds of cylinders here in our works, that very few cylinders will clean out with less than ten or twelve-thousandths of an inch of stock removed.

When, however, the scoring is deep and the cylinder shows a great deal of wear twenty-five or thirty or even forty thousandths of an inch have sometimes been removed in grinding before a satisfactory surface and hole accurately round and straight can be produced.

It is the writer's impression, therefore, that two oversize standards would probably be sufficient in order to reduce to the lowest terms the stock that any agent would be obliged to carry, and his idea is that fifteen-thousandths larger for the first oversize standard and thirty-five or forty-thousandths for the second would really be found ample for all cases and a most satisfactory combination.

The fifteen-thousandths increase would practically be 1-64 inch larger than standard, while the forty-thousandths increase is only 1-25 inch. There is no question but what any cylinder in which the walls are of reasonable thickness will stand this amount of increase; in fact, a great deal more, as this amounts to but 1-50 inch on each side.

**THE ORDINARY REPAIR MAN**—The first argument against this scheme will probably be the difficulty of getting the cylinders refinished or reground by the ordinary repair man.

With regard to this point, I would say that this is a simpler matter than at first appears; because a number of repair men in different cities have already equipped their shops with cylinder grinding machines, so that they are prepared to regrind a cylinder at almost a moment's notice, and only lack the adoption of these oversize standards by the manufacturer to be in position to make the most rapid and economical repairs possible.

While some of the repair shops have installed new cylinder grinding machines purchased directly of the machine tool manufacturer, many of the repairs shops have felt that the cost of a new machine was higher than they could afford, and have solved the problem by installing second-hand machines.

These second-hand cylinder grinders can often be secured through machinery dealers in different sections of the country, who have taken them in trade from the automobile manufacturers for machines of later type and containing the very latest and most modern improvements.

The engine builder, if he is manufacturing his motors in the most efficient manner, will wish to have in his equipment the most modern and improved grinding machinery possible to find, and as great improvements in this type of machinery have been made of late it is simply good policy for him to trade

off his older machines, which have already paid for themselves in excellent service, and have his grinding equipment of the most modern character possible to secure.

These used machines, however, will be found to be thoroughly satisfactory to the average repair man for the amount of grinding he will have occasion to do and they can usually be secured for a small portion of the original cost, say from one-third to one-half of the price of a new machine.

This enables the repair man who appreciates the value of one of these machines to secure it with a moderate investment, and the returns on this investment will be found to be extremely satisfactory.

It is not necessary, however, for every repair man to invest in a cylinder grinder to make this plan a success, as a single machine, even in a city of moderate size, would probably be sufficient to take care of all the work of this kind needed in that vicinity, and enable all those who were making repairs to profit by this improved method of handling the work.

This changing off of machines is also an excellent thing for the manufacturer, as it enables him to keep his equipment in

the best possible condition for manufacturing at the lowest cost consistent with high-grade work.

It therefore seems to the writer that these two factors, the manufacturer and the repair-man, can work together along these lines to the greatest possible good to themselves as well as to the purchaser of automobiles.

**CO-OPERATION OF MANUFACTURER AND REPAIR-MAN NECESSARY**—Now this whole proposition to be most satisfactory must be mutual; it is of no great advantage for the repair-man to install a cylinder grinder for the refinishing of holes and still be obliged to go to the job shop for his pistons and rings; neither are the full advantages secured for the manufacturer to adopt and furnish the oversize pistons and rings unless the cylinders can be readily refinished to oversize dimensions to accommodate these. But the advantages are so obvious and the means of accomplishing them are so easy that it is the writer's most earnest hope that this will be appreciated by those concerned, and this most desirable arrangement be worked out to the mutual advantage of all parties, both financially and in the matter of time, in making engine repairs involving cylinders, pistons and rings.

## Designs of Aluminum Castings

By W. H. Gilbert, Member S. A. E.

*Paper read at the summer meeting of the Society of Automobile Engineers, at Dayton, O., June 15-17, 1911. The author considers the subject from the viewpoint of the designer and the pattern maker, and discusses among other things the subjects of contraction on freezing, pouring temperature, thickness of section, alloys, endurance, and the designing engineer's responsibility.*

**T**HE lightness, beauty and resistance of corrosion of cast aluminum, combined with its good strength in comparison with its weight, are bringing it into vastly increased use in the best modern cars. All automobile castings must, of course, be designed primarily with a view to the fitness for the particular use intended. Usually, however, there are several ways of designing an aluminum casting, any one of which would allow it to serve its purpose, keep within the weight limit and be a good proposition from the point of view of the machine shop.

**THE PROBLEM**—The problem is then to select the best possible design so that the casting may be produced in the best and cheapest way. Luckily the design that allows a casting to be easily and cheaply made is usually one that gives us a casting of far greater strength and reliability than one so designed that it is a difficult one to handle in the foundry. The designer is usually closely in touch with the machine shop and its problems, but seldom is he a foundry expert; could he realize the extra labor,

and hence the extra cost and delay in production, a seemingly slight point in his design may make in the foundry, he would more often consult with the pattern maker and with the foundryman before completing his design.

**THE DESIGNER AND THE PATTERN MAKER**

—Owing to certain physical properties of aluminum, such as its high contraction on cooling and its weakness when just solidified that is, its hot shortness—aluminum castings require more careful design than almost any other casting metal. If one examines the defective casting records of the individual patterns in a large aluminum foundry he is struck at once by the vast difference in the results from different patterns. Some patterns give uniformly good results, while others differing but slightly from them are extremely troublesome. Speaking broadly, almost 50 per cent. of the discrepancies between the number of molds put up and the number of good castings made may be traced to the door of the designer in one way or another, and about 25 per cent. more may be traced to the door of the pattern maker.

**COST AND DELIVERY**—Do not think that we look upon a designer and a pattern maker as the natural enemies of the foundry. On the other hand, they are keen to see that what they can do to aid the foundry without interference with the requirements which the casting must meet, will come back to them many fold in lower cost and in regularity of delivery. But it so often happens that the automobile engineer will say: "We know that particular design is a poor foundry proposition and that pattern is not made in the best possible way, but we cannot change that model now nor can we tie up production of our cars long enough to allow changing the pattern," that it is worth while to remind you that the time to go into these points is when a new model exists in the mind and on the drawing paper of the engineer, rather than when the car is being assembled.

**CONTRACTION ON FREEZING**—In passing from the molten to the solid state aluminum contracts a good deal; when a heavy and a thin section come next to each other, as in sample No. 1 (which will be passed around), the thin place will freeze first. If the thin section is so situated as to lie between a heavy section and a gate or riser the supply of metal is thereby cut off from the molten mass in what is to be the heavy part of the casting. The contraction on freezing has to take place, and instead of taking place uniformly over this heavy part and maintaining the exact shape of the mold, it will often draw away from a corner and produce a "shrink," as in this sample. We can induce the heavy portion to freeze more quickly by placing the chill in the mold

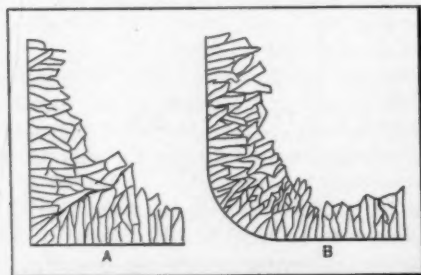


Fig. 1—(A) showing formation of crystals in the cooling of a sharp-cornered casting. (B) on a smooth curve

at that point (a chill being a piece of some material, usually a metal having a higher heat conductivity than the pot of the mold). It is difficult to accomplish the end completely by this method; it greatly increases the time required to put up the mold and produces unsightly chill marks on the casting.

The ideal casting, therefore, is one of as nearly uniform section throughout as is practical, since that means that the whole casting solidifies at the same time, so that contraction is uniform. On account of the hot shortness of aluminum the shrinkage strains set up when a heavy section joins a thin one often causes the metal to give away entirely at that point, and a crack appears, as in sample No. 2. Sample 3-A shows this even better. This was taken from a crank-case containing three braces—two like 3-A and one like 3-B. Those of the 3-A type cracked in a consistent and appalling manner, while the 3-B never gave any trouble. It was finally found that there was no reason at all why all the braces should not be like 3-B, and the designer finally gave permission to have them all made in that way. The defective castings from that cause promptly fell from fully 30 per cent. to nothing. This change meant simply the addition of an ounce of metal to a 48-pound casting. Moreover, this principle can as often be applied by reducing too thick a section as by increasing too thin a one.

**FILLET**—If it is inevitable that light and heavy sections come together, the cooling strain should be distributed by joining the sections by a smooth curve, that is, a liberal fillet. This is on account of another physical property common to all molten metals. Suppose we have a sharp corner, as at the vertex of a right dihedral angle. The metal, of course, freezes first at the edge, crystals being deposited which tend to grow inward, as shown in Fig. 1-A. Each succeeding crystal finds it easier to attach itself to the end of the one previously formed, so that soon a line of crystals has grown by bisecting the angles. Now crystals in regular lines do not form as strong a mass as when they lie interlaced in helter-skelter fashion. It is easier to pull apart a pile of nails lying side by side than one which has been stirred up. On a smooth curve, as in Fig. 1-B, there is no adventurous crystal which comes out first and to which the rest attach themselves like a swarm of bees. The crystals among the curve surface come out more nearly at the same time, and the next lot is deposited lying between the first ones in helter-skelter fashion, instead of coming out in a line with the "follow-my-leader" effect. In other words, a sharp angle tends toward the formation of a definite line of crystals, which forms a sort of cleavage plane and is a source of weakness.

A break across a square bar of almost any metal will show distinct lines connecting the opposite corners, showing the arrangement of the crystals in this fashion, while a round bar shows no such lines. From this we can see the necessity for a liberal fillet. The sharper the angle at a change of direction, or the greater the difference of thickness at a change of section, the greater should be the radius of curvature of the fillet. If a second heavy part has a large boss and is near a first, both being joined by a thin section, the fillet should be more liberal than if only one were present. Sample 4 is an example of the lack of fillet at a sharp corner between thick and thin sections. This was the fault of the pattern maker in this instance.

**POURING TEMPERATURE**—There is no one factor in foundry practice that more gravely affects the strength of the casting than the pouring temperature. The reason for this is again the speed of crystallization. The cooler the metal can be poured into the mold the more quickly it solidifies and the less time the crystals have to grow or arrange themselves, and the result is a mass of closely interlocking crystals forming a strong fine-grained material.

The effect of pouring temperature is well shown by this set of test bars, all of which were cast from the same pot of metal with exactly similar molds, the only variable being the pouring temperature. The bars are arranged in the order of pouring temperature, as shown by the label on the other side, which also gives the strengths of the individual bars. You will note

the rough surface and coarse fracture of the bars cast at the higher temperatures, as well as their low strength. The average results obtained in this series of tests are given by the curve seen in Fig. 2, which shows that the lower the pouring temperature the stronger the casting.

**THICKNESS OF SECTION**—This has a distinct bearing on design, since the lowest temperature at which

a casting can be poured is that to which the thinnest section will just escape a misrun. If the casting is so designed that this crucial section forces you to pour hot, all of the thicker parts will freeze too slowly and will be weaker than they should be. By slightly increasing the section of the thinnest parts, a casting could often be poured 100 degrees colder and the strength of the whole casting be increased at least 10 per cent. This is a matter which is absolutely up to the designer. If the bulk of the casting is from a quarter to a half inch thick, one little part one-eighth of an inch thick will give us a resultant casting, on account of the high pouring temperature required, whose average strength is about 16,000 pounds per square inch, instead of 18,000 pounds or over. The call for lightness has led many designers to overlook this vital point. We believe that an average crankcase should not contain any section thinner than one-quarter inch.

**TEST BARS ON CASTINGS**—The very great influence of the pouring temperature is the reason why separately cast test bars show only the quality of the ingot metal and nothing at all as to the strength of the corresponding casting, even though the test bar and casting may be poured from the same pot of metal. Your Standards Subcommittee has wisely specified that aluminum test bars shall be made on castings. Were this stipulation not made the foundryman who wished could pour the casting as hot as he pleases, allow his metal to cool away down and then pour separate test bars which would then show an utterly fictitious strength in comparison with the casting.

**POURING TEMPERATURE AND PUBLISHED DATA**—The general lack of attention to pouring temperatures, not only in commercial practice but in most of the investigations on aluminum, vitiates a great deal of the published data on aluminum alloys and accounts for a great many irregularities and seeming contradictions in the results. In comparing the different aluminum alloys, really comparable results can only be obtained by pouring at the same number of degrees above the melting point of the particular alloy in question in all cases, thus allowing the same time for crystallization and producing an analogous condition.

**THE PATTERN**—After the designer has done his work it is up to the pattern maker to decide how the pattern shall be constructed. First of all, the pattern should be so made as to allow the use of molding machines wherever practical. If a pattern-maker is not too much bound down by tradition he can often simplify matters greatly. For instance, in a crankcase with several projecting pieces on the side which would make it impossible to draw the pattern from the sand without the use of core work at the sides, by the simple expedient of making the pattern hollow and putting in a lever by which the pieces are drawn into the body of the pattern and the pattern then lifted out, a large amount of core work was eliminated.

**CORE WORK**—Core work always means trouble. It takes time to set cores in the mold correctly, and if a lot of small cores are used the danger of shifts is greatly increased. If on the other hand large cores are used they must be made hard enough

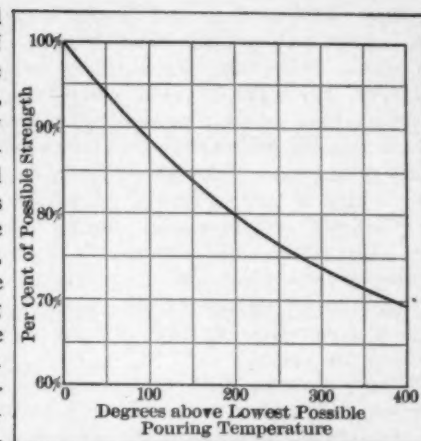


Fig. 2—Curve showing results obtained in casting aluminum at varying temperatures

to allow handling them and setting them in the mold, which requires not only a solid core but one reinforced by iron rods and wires. This makes them hard to crush, and on large cores inside of thin walls of metal introduces danger of cracking. When we have a core completely surrounded by walls of metal it is a question whether the tensile strength of the metal as it solidifies is greater than the compressive strength of the core. Let the core be ever so slightly too hard and your casting is inevitably scrap. We have seen patterns requiring large and complex cores within walls as thin as 3-16 of an inch, from which patterns it was absolutely impossible to make castings, since any core strong enough to place in the mold would have too great a compressive strength for the metal to crush without cracking the casting.

If cores must be used, the core prints should be large and deep so as to anchor the cores firmly without the use of chaplets to hold the cores in place, since it is impossible for the molten metal to fuse a chaplet into body of the casting, without pouring at a temperature far above that necessary to give the greatest strength. When a job requires cores, the first question that should be asked by the pattern maker is if that pattern cannot be made so as to allow the use of green sand core, or at least a green sand half. Green sand will crush and give way when the casting contracts on cooling, where a hard, dry sand core will not crush and will crack the casting.

The foundry is called upon to make castings of ever greater complexity, and the core work required has increased amazingly in the last couple of years. Core rooms once fully adequate for a given molding room are now being overcrowded and unable to cope with the demand for cores. This situation could be greatly improved if the pattern maker would put green sand halves in all castings where it is possible to use them; and the results would be both cheaper and better.

**METAL PATTERNS AND CORE BOXES ECONOMICAL**—If a considerable number of castings are to be made, wooden patterns and core boxes are an abomination. They warp and swell in use, dry out and crack apart in storage and wear out by abrasion from the sand and by the constant rapping necessary to allow the pattern to be withdrawn from the mold or the core box from the core. Good metal patterns and core boxes are an economy if many castings are to be made, or if a few are to be made from the same pattern in successive years.

**ALLOYS**—With proper design and a well-made pattern the engineer who wants an aluminum casting must next face the problem as to what aluminum alloy he shall specify his casting to be made from. He has a choice of practically three alloys—one containing 8 per cent. copper, one with 3 per cent. copper and 15 per cent. zinc, and one with 35 per cent. zinc, designated respectively as alloys Nos. 1, 2 and 3 in the Standards Subcommittee specifications and known to the trade as Nos. 12, 31 and 63. Comparable test bars on these show in the order named—18,000 pounds, 22,000 pounds and 35,000 pounds tensile strength per square inch.

Probably 90 per cent. of all aluminum automobile castings are made of No. 12. Though this shows the lowest tensile strength of the three, the trade has come to it for several reasons. First, when aluminum castings are desired lightness is usually a prime factor and the zinc-containing alloys, Nos. 31 and 63, are respectively about 7 per cent. and 18 per cent. heavier than the copper containing alloy, or No. 12. The second reason is that the alloys containing zinc are more liable to shrinkage strains, which may develop as draws and shrinks or as hidden strains, which cause cracks to appear after the material has been subjected to vibratory stress, and make the casting liable to fail in service. The strength of the zinc alloys falls off more rapidly with increasing temperature than does that of No. 12, so that the alloys which are stronger at ordinary temperatures, when subjected to the heat developed about a motor, may lose their advantage.

**ENDURANCE**—The zinc alloys, moreover, are more brittle and less ductile than the No. 12, as well as more likely to break down in vibration or under repeated impact. Although so eminent an

authority as Mr. Souther disagrees with this in a recent article in the *Cycle and Automobile Trade Journal*, yet there must be a reason for the very marked abandonment of the zinc alloys for the copper alloy throughout the industry. Mr. Souther's figures given in this paper are the same he published several years ago in the *Metal Industry*, and on looking over the fuller data in this earlier paper it seems to us he must have drawn conclusions from too few tests and probably from tests on bars not necessarily cast at corresponding temperatures.

We expect to go into this matter of endurance of different aluminum alloys more fully in the near future and plan to report the results at a future meeting of the society. A large number of tests so far have shown that on the White-Souther endurance machine the different alloys run about as follows:

No. 12—over a million revolutions;

No. 31—about 600,000 revolutions;

No. 63—about 500,000 revolutions,

before fracture. These are averages of a large number of bars. You will note that the resistance to vibratory stress is in opposite order from that of the tensile strength, but exactly in the order of the ductilities.

On another type of endurance machine where the bar is subjected to repeated blows, we understand the No. 12 will stand about a million blows before rupture, while No. 31 will only stand 1,500 to 2,000.

The brittleness and unreliability of No. 31 and No. 63 is usually laid to the presence of zinc. On the other hand, it is more likely due to the absence of aluminum—that is, alloys high in aluminum are resistant to vibratory stresses, while alloys low in aluminum are less resistant. An alloy containing 88 per cent. aluminum and the rest zinc gives nearly as good results in the endurance test as does No. 12 with 92 per cent. of aluminum, but since the tensile strength is decidedly inferior to No. 12, and it is a heavier alloy, this is no longer a commercial alloy.

Magnalium, or 94 per cent. aluminum with 6 per cent. magnesium, is about as strong in tension and has about the same resistance to vibratory stress as has No. 12. To get strength enough to pay for the added weight where zinc is used as the chief alloying metal, we are forced to use so much zinc that the ductility and resistance to vibratory stress are cut far below the figure for No. 12. We really have three factors, strength, lightness and resistance to vibratory stress. If we give each factor what we might call a coefficient of importance and multiply the three together, the alloy which gives the highest product will be the most valuable.

The automobile engineer has practically made this calculation and he has answered the question by specifying the 8 per cent. copper alloy in the vast majority of cases.

**THE DESIGNING ENGINEER'S RESPONSIBILITY**—The engineer who wishes to get good reliable aluminum castings and at a reasonable price must then realize that the problem is not entirely up to the foundryman to whom he entrusts the handling, but that it is his problem as well. The responsibility for design is solely his, and if he will so proportion his castings as to avoid great inequalities in section or sharp changes in direction, and if he will use liberal fillets, if he will keep away from such thin sections that the metal must be poured too hot to get strong castings, if he will instruct his pattern maker to go as far as he can in avoiding dry sand cores, if he will provide good substantial patterns, will specify the proper alloy and then see that the casting is made with proper care and with proper control of pouring temperatures, he will get castings which will allow him to utilize to the fullest extent all the many advantages which aluminum has for his purposes. But he should not for one moment forget that unless the design is right the best efforts of the foundryman cannot produce a casting possessing in the fullest degree those advantages which the perfect aluminum casting should have.

The designer is father to the blue print—the blue print is father to the pattern—and the pattern is father to the casting. Even though the relationship be remote, the designer should give his great-grand-children the best possible chance.

# Vibration in Gasoline Motors

By A. Cattaneo

The following paper, which was first published in the *Motor Trader*, London, discusses the problem of motor balance and the disturbing moments, analyzing the conditions prevalent in four-cycle engine practice which have bearing upon this class of phenomena, reaching, by this process, a number of conclusions of interest to gasoline engine designers and users. The treatise sets forth, in a clear and vivid manner, the varying conditions during the four strokes, and their effects upon motor balance, which is an item in engine efficiency and depreciation.

**SPEED VS. VIBRATION.**—The balance of a well-made four-cylinder engine is generally considered quite satisfactory, although one almost instinctively feels that it is not as perfect as it is often claimed to be.

That the balance of these engines is not altogether perfect is clearly shown by the circumstance that, while a motor runs steady under normal conditions, it develops a decided tendency to vibrate as the speed increases; the vibrations becoming objectionable when the engine races beyond a certain number of revolutions. It is obvious that speed alone cannot be held responsible for any vibration since the latter represents actual "work," which, we know very well, only obtains through the simultaneous existence of both speed and force. In our case, where does the latter come from, how does it give rise to the disturbance already referred to? I found it most difficult to trace the origin of the disturbing force or forces through the formidable display of complicated formulae deemed necessary for an adequate study of the problem. To know that the "primary" or "tertiary" forces are balanced or unbalanced is of little interest, and of no use whatever to us, unless it is previously made quite plain how such forces exist and act, and the meaning of such designations as "primary," "secondary," etc., defined, not in view of the mathematician's convenience, but for that of the practical engineer, to whom clearness, simplicity and reliability have a paramount importance.

There is a peculiarity about the usual crank and connecting rod mechanism which constitutes a latent cause of vibration not to be avoided unless a radical change in design is resorted to. The point is important and interesting enough to command the attention of engineers.

This will be dealt with as clearly as possible in the course of this article.

**DOWN-STROKE VIBRATION.**—Referring to Fig. 1, which represents an engine with its piston traveling on a down stroke, let  $x$  be the piston travel measured from the top dead center,  $\beta$  and  $\alpha$  the corresponding angles of the connecting rod and crank respectively. Then we have:

$$x = l + r - l \cos \beta - r \cos \alpha$$

But it also is

$$b = l \sin \beta \text{ and } b = r \sin \alpha \text{ hence}$$

$$\sin \beta = \frac{r \sin \alpha}{l}$$

on the other hand:

$$\cos \beta = \sqrt{1 - (\sin \beta)^2} = \sqrt{1 - \left(\frac{r \sin \alpha}{l}\right)^2}$$

Before this value for  $\cos \beta$  is introduced in the equation of the piston travel it will be found advantageous to eliminate the square root, by developing it in a series, according to the Binomial Theorem. We have then:

$$\cos \beta = \sqrt{1 - \left(\frac{r \times \sin \alpha}{l}\right)^2} = \left[1 - \left(\frac{r \times \sin \alpha}{l}\right)^2\right]^{\frac{1}{2}} = 1 - \frac{1}{2} \times \left(\frac{r \times \sin \alpha}{l}\right)^2 + \frac{1}{2} \times \frac{1}{2} \times \frac{1}{2} \times \frac{1}{2} \times \frac{1}{2} \times \left(\frac{r \times \sin \alpha}{l}\right)^4 - \dots$$

It will be seen that the third term of the above series is negligibly small as compared with the second and can therefore be left out of consideration.

The formula for the piston travel then becomes:

$$x = l \times r - l \left[1 - \frac{1}{2} \times \left(\frac{r \times \sin \alpha}{l}\right)^2\right] - r \times \cos \alpha = r(1 - \cos \alpha) + l \times \frac{1}{2} \times \left(\frac{r \times \sin \alpha}{l}\right)^2$$

If  $s$  is the ratio  $r/l$  of crank radius to connecting rod length, the above formula becomes:

$$x = r \left[1 - \cos \alpha + \frac{1}{2} \times s \times (\sin \alpha)^2\right]$$

**UP-STROKE VIBRATION.**—The case of the return stroke is shown in dotted lines in the same diagram (Fig. 1); the angle  $\alpha$  of the crank-pin, and consequently the angle  $\beta$  remaining the same as before, and calling again  $x$  the piston travel measured from the lower dead center, we find:

$$x = l \times \cos \beta - (l - r) - r \times \cos \alpha$$

we found already that

$$\cos \beta = 1 - \frac{1}{2} \times \left(\frac{r \times \sin \alpha}{l}\right)^2$$

hence

$$x = r \left[1 - \cos \alpha - \frac{1}{2} \times s \times (\sin \alpha)^2\right]$$

The general equation of the piston travel expressed as a function of the crank-pin angle is therefore:

$$x = r \left[1 - \cos \alpha \pm \frac{1}{2} \times s \times (\sin \alpha)^2\right]$$

wherein the  $+$  and  $-$  signs refer to a down and up-stroke respectively.

**A DEDUCTION BY ANALYSIS.**—It will be seen, therefore, that, with a given radius of crank, connecting rod length and crank-pin angle, the piston travel is greater for a down than for the opposite stroke. This apparently paradoxical result is simply due

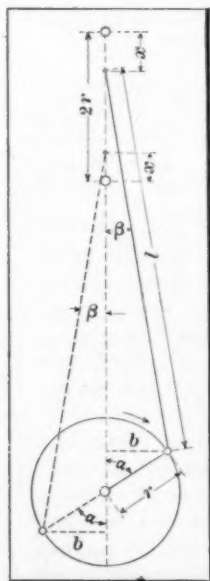


Fig. 1—Diagram of engine with piston traveling on the down stroke

to the fact that the ratio is not infinitely great, but is limited by construction requirements to a value which very seldom exceeds 5/1. If  $l/r = \infty$ ,  $s$  becomes 0, and the travel is  $x = r(1 - \cos \alpha)$ .

It may be interesting to know the value of the difference between the travels of the piston for two opposite strokes. Assuming  $r = 2$ ,  $l = 7$ , and  $\alpha = 30$  deg., the piston travel  $x$  itself becomes:

$$x = 2 \left[ 1 - .866 + \frac{1}{2} \times \frac{2}{7} \times .25 \right] = .34$$

and

$$x = 2 \left[ 1 - .866 - \frac{1}{2} \times \frac{2}{7} \times .25 \right] = 1.96$$

for a down and up-stroke respectively; that is to say, the piston travels are, in this case, in the ratio of about 2 to 1.

**CONSTANCY OF CRANK SPEED.**—On the other hand, we know that the crank travels at constant speed. Therefore, the time required for describing the angle  $\alpha$  is invariably the same, no matter from which dead center this angle is measured.

The times being the same, and the piston travels different, it naturally follows that the piston velocities must also be different, according to whether the piston is on its down or up-stroke. The piston velocity  $v$  can be very easily calculated from the formula for  $x$ . It is, namely:

$$v = -r \left( \sin \alpha \pm \frac{1}{2} \frac{da}{dt} \times \cos \alpha \right).$$

On the other hand — is the angular velocity of the crank-pin; the

speed  $w$  of the latter is therefore  $w = \frac{da}{dt} \times r$ .

consequently

$$v = w \cdot \sin \alpha (1 \pm s \times \cos \alpha).$$

I will refer again to this formula when discussing the relation between the average and maximum piston speed, concerning which erroneous statements are often made and sometimes also in print.

**PISTON VELOCITY.**—We have found that with a given crank-pin angle the velocity of the piston is greater on a down than on the up-stroke. It obviously follows that the acceleration  $a$  of the piston is also greater in the former case. On the other hand we know that in order to give a mass  $m$  an acceleration  $a$ , a force  $f$  is required, and that between these quantities the fundamental relationship invariably exists.

$$f = m \times a.$$

In our case the mass  $m$  (of the piston and fraction of connecting rod weight) is constant, while the acceleration  $a$  varies. The force  $f$  which produces this acceleration must therefore also be different, namely, greater and smaller on a down and up-stroke respectively. In order to calculate this force  $f$  we must know the value of the acceleration  $a$ ; the latter can be easily deduced from the above formula.

$$a = -w \cdot \cos \alpha \pm \frac{1}{2} w \times s (\cos^2 \alpha) \pm \frac{1}{2} w \times s (\sin^2 \alpha) - \frac{da}{dt}$$

We found that  $-r = w$ , therefore  $\frac{da}{dt} = \frac{dw}{dt} \times r$ . By substituting the value in the above formula we obtain:

$$a = -\frac{w}{r} \left[ \cos \alpha \pm s (\cos^2 \alpha) \pm s (\sin^2 \alpha) \right]$$

But  $2(\cos^2 \alpha) = 1 - \cos 2\alpha$ , and also  $(\sin^2 \alpha) = 1 - (\cos^2 \alpha)$ . Hence the equation for  $a$  becomes:

$$a = -\frac{w}{r} \left[ \cos \alpha \pm s (\cos^2 \alpha) \pm (1 - \cos^2 \alpha) \right] = -\frac{w}{r}$$

$$\left[ \cos \alpha \pm s (2 \cos^2 \alpha - 1) \right]$$

and finally  $a = -\frac{w^2}{r} (\cos \alpha \pm s \times \cos 2\alpha)$ .

The corresponding accelerating force  $f$  is then:

$$f = m \times \frac{w^2}{r} (\cos \alpha \pm s \times \cos 2\alpha).$$

Generally the number of revolutions of the crank is known. It is namely:

$$w = \frac{2 \times \pi \times r \times n}{60}$$

hence:

$$f = m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (\cos \alpha \pm s \times \cos 2\alpha).$$

**THE FORCE "F."**—This force  $f$  varies in value and sign during a complete stroke. It is a maximum and positive at the beginning of a down-stroke, and diminishes rapidly until it becomes zero at that point of the stroke for which the piston velocity  $v$  already referred to is a maximum. From this point it increases again in absolute value, but becomes negative, or in other words, it becomes a retarding force. A similar remark also applies in case of the return stroke.

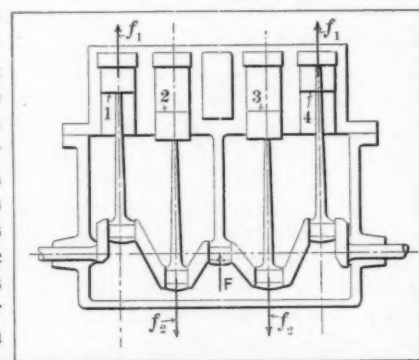


Fig. 2—Illustrating forces in a standard type four-cylinder motor

The most important values of the force  $f$  are given below, and are simply obtained by making the angle  $\alpha$  zero or 180 deg., according to the dead centers and strokes considered. It must also be noticed that the retarding force is opposite in direction to the accelerating force, hence if the latter is positive, the former must be negative, and be accordingly affected with the minus sign. A down-stroke begins with an accelerating force:

$$f = m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 + s).$$

and finishes with a retarding force:

$$f = m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 - s).$$

The return stroke begins with the accelerating force:

$$f = m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 - s).$$

and closes with a retarding force:

$$f = m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 + s).$$

**APPLICATION OF ANALYSIS.**—In the four-cylinder engine built according to the usual practice and shown in Fig. 2, the reciprocating masses are the same for each cylinder, and assuming the pairs of pistons 1 and 4, and 2 and 3, to be at the end of their return and down strokes respectively, we obviously find that the crankshaft is acted upon by two forces, namely,  $2 \times f_1$  and  $2 \times f_2$  acting in opposite directions. The arrangement being quite symmetrical, these forces may be considered as acting in the central vertical plane of the engine. It is, therefore, quite

clear that were these two forces equal they would neutralize each other and the engine would, in this respect, be automatically balanced. These two forces differ in absolute value; we have found that a return stroke closes with a retarding force:

$$f = m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 + s).$$

Therefore the pistons 1 and 4 of Fig. 2 are cumulatively responsible for a force:

$$2 \times f_1 = 2 \times m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 + s)$$

acting in an upward direction, while the remaining pair of pistons, 2 and 3, are responsible for a force:

$$2 \times f_2 = 2 \times m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 - s)$$

acting in a downward direction.

The difference  $2 \times f_1 - 2 \times f_2 = F$  is a free force acting in the central vertical plane of the engine and—for the case considered—in the direction of  $f_1$ , namely, upward.

Similarly, we would find, by referring to what has been previously said, that when both pistons 1 and 4 are starting their down strokes, the difference  $2 \times f_1 - 2 \times f_2 = F$  remains constant, what absolute value is concerned, but changes its sign, that is to say, the resulting force  $F$  acts now in a downward direction. The force  $F$  itself is obviously:

$$F = 2 \times f_1 - 2 \times f_2 = 2 \times m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 + s) - 2 \times m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times (1 - s) = 4 \times m \times \left( \frac{\pi \times n}{30} \right)^2 \times r \times s.$$

**UNBALANCED FORCE**—It will be interesting to know the exact value of this unbalanced force in case of an engine of current dimensions. For this purpose we will assume the reciprocating weight of a piston, plus one-third of that of a connecting rod, to be 4 kilos (8.45 pounds), which is approximately the average for a 30-horsepower, four-cylinder engine. With a crankshaft speed of 1,200 revolutions per minute, a ratio  $r/l = 1/5$ , and a radius  $r$  of crank of 6 centimeters, the force  $F$  becomes:

$$F = 4 \times \frac{4}{10} \times \left( \frac{3.14 \times 1200}{30} \right)^2 \times \frac{1}{5} \times .06 \times \frac{1}{5} = 275 \text{ kilos (605 pounds).}$$

It will be readily perceived that this force, acting alternatively in an up and downward direction, may be responsible for a certain amount of vibration, which, however, on account of the comparatively heavy mass of the engine, and also on account of the circumstance that the force acts in a vertical plane, is not necessarily objectionable as long as the motor is properly suspended and its number of revolutions per minute kept within reasonable limits.

**VERTICAL vs. HORIZONTAL CYLINDERS**—While it is impossible to eliminate the disturbance above referred to on vertical engines, the difficulty is very easily got over in case of horizontal motors with opposed cylinders. The writer only knows of a single instance where this advantage was fully recognized, and the engine accordingly designed. I refer to the Oerlikon motor (cf. "Trader" issue, Dec. 7th ult., p. 1,387 and Fig. 3). Referring to the latter, let 1, 2 and 3, 4 be the set of cylinders on the right and left respectively when the engine is viewed from the flywheel, cylinders 1 and 3 being nearest the latter. If pistons 1

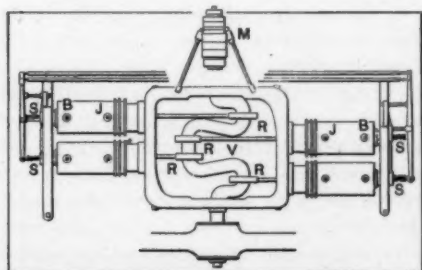


Fig. 3.—The Oerlikon four-cylinder, three-crank motor

and 3, and pistons 2 and 4 are at the end of a return and "down" stroke respectively, it is evident—according to what I have demonstrated—that the retarding forces of pistons 1 and 3 are equal to each other, opposite in direction, and greater than the retarding forces due to pistons 2 and 4; which forces are also equal to each other, although of opposite directions. The shaft is accordingly acted upon by two sets of forces, namely,  $f_1$  and  $f_4$  in one direction (right) and  $f_2$  and  $f_3$  in the other direction (left). Since  $f_1 = f_3$ , and  $f_2 = f_4$ , and since the distance between the axis of any two cylinders is constant, the resulting couple or force  $F$  is zero; that is to say, the engine is perfectly balanced in this respect. This easily accounts for the astonishingly smooth running, even at comparatively high speed, of the Oerlikon engine, for which lightness and an exceedingly long stroke (4 inches by 8 inches) add further difficulties against the elimination of vibrations.

**INCONSTANCY OF PISTON VELOCITY**—Another class of vibration in gasoline engines is also to be traced to the unsymmetrical travel rate of the piston referred to in the early part of this article. We will illustrate it only for a point of the stroke, namely, for the point for which the piston velocity  $v$  is maximum. Referring to Fig. 4, let the latter represent the end view of the four-cylinder engine shown in Fig. 2, the crankshaft having so far revolved that both pistons 1 and 4 have attained their maximum speed velocity. For this particular point the acceleration of the pistons just named is zero. Assuming no static pressure is acting on these pistons, it will be readily seen that, notwithstanding the angularity of the connecting rod, no side thrust is exerted on cylinders 1 and 4, since no force is at that particular moment available on the piston. In order that same conditions may obtain for pistons 2 and 3, their cranks need have reached the point  $x$ . Instead of this, however, they obviously find themselves at the point  $y$ , that is to say, the pair of pistons just referred to have not yet reached the point of the stroke for which the piston speed is a maximum. A certain amount of accelerating force is consequently still acting on this pair of pistons, and this force is obviously responsible for the corresponding amount of side thrust which is not balanced by an equal and opposite thrust on the remaining cylinders.

Here again the disturbance need not be great within a moderate range of speed; but it must not be forgotten that this thrust, as well as the force  $F$  previously referred to, vary as and with the acceleration, which, in turn, is proportional to the square of the number of revolutions, as shown by the formula for  $a$ . That is to say, if the speed increases from 1,000 to 2,000 revolutions per minute, the acceleration becomes four times greater, and under these conditions the vibrations may easily become objectionable.

**POINTS OF MAXIMUM PISTON VELOCITY**—The formula for the piston velocity is:

$$v = w \times \sin^a (1 \pm s \times \cos^a).$$

Let us first of all find for which values of the angle  $a$  this velocity is a maximum. In the case of a down stroke we have:

$$\frac{dv}{da} = w [\cos a \times da + s (\cos a)^2 da - s (\sin a)^2 da] = 0$$

or

$$2 \times s \times (\cos a)^2 + \cos a - s = 0 \quad \cos a = \frac{1}{4 \times s} \pm$$

$$\sqrt{\left( \frac{1}{4 \times s} \right)^2 + \frac{1}{2}}$$

Of the two solutions, the one which corresponds to the sign

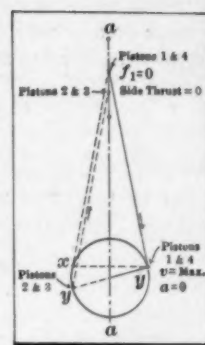


Fig. 4.—Diagram representing end view of four-cylinder motor shown in Fig. 3.

+ needs alone to be considered, since the other gives for  $\cos a$  a value greater than one, which we know is not admissible. Consequently, assuming again a ratio  $r/l = 1/5$ , we find:

$$\cos a = -\frac{1 \times 5}{4 \times 1} + \sqrt{\frac{25}{16} - \frac{1}{4}} = .185$$

therefore  $a = 79$  degrees.

For an up stroke we should find in a similar manner  $a = 101$  degrees. The corresponding piston travel  $x$  is:

$$x = r(1 - \cos a \pm s \times \frac{1}{2} \sin^2 a) =$$

$$r(1 - .185 \pm \frac{1}{5} \times \frac{1}{2} \times .98^2) = r \times .90$$

$$\text{and } r = l \times 1.1$$

for a down and up stroke respectively.

It will therefore be seen that the maximum piston velocity obtains at about 9-20ths of the stroke from the top dead center in case of a down stroke, and at 11-20ths of the stroke from the lower dead center for the return stroke.

The actual value of the maximum velocity itself is obviously:

$$v = w \sin a (1 + s \times \cos a) = w \times .98 (1 + \frac{1}{5} \times .185) =$$

$$w \times .98 \times 1.03 = w \text{ app.}$$

In other words, with the usual ratio of crank radius to connecting rod length, the maximum piston velocity is practically equal the constant velocity of the crankpin.

On the other hand, we know that the mean speed  $W$  of the piston is:

$$W = \frac{4 \times r \times n}{60} = \frac{r \times n}{15}$$

and the crankpin speed.

$$w = \frac{\pi \times r \times n}{30}$$

Hence

$$\frac{w}{W} = \frac{\frac{\pi \times r \times n}{30}}{\frac{r \times n}{15}} = \frac{\pi}{2} = 1.6 \text{ or } w = 1.6 \times W.$$

That is to say, the maximum velocity of the piston is 1.6 times greater than the mean piston speed.

THE most important department of a business should not be left to a man whose sole interest lies in a commission on the value of the space that he can fill with words.

## Voiced by the Captains

### Shedding Light on Intricate Phases of Automobiling

*Timely editorials on current subjects involving the automobile by well-known heads among those of the industry, touching upon the ideas that suggest an extra measure of success, if they are properly handled, and showing how the engineers may do more and better work with perhaps less effort.*

WHILE the engineers of the automobile industry are sojourning at Dayton for the purpose of attending the Summer Meeting of the Society of Automobile Engineers, they will be in a receptive mood in relation to the promising situations, and they may have a few moments to spare which may be profitably put in, taking note of what other men think, thereby broadening their perspective. There are quite a number of matters that should command the attention of the thinking engineers, and the aim has been in the editorials here presented to open up these branches in the general line of thought.

### Reform Faulty Designs Rather than Pay Fines

President Benjamin Briscoe, of the United States Motors Company, enters a brief in favor of the engineer, asking him to be considerate of the user, telling him to correct his faults by changing the design, in a given case, if necessary, rather than to try to circumvent the trouble by resorting to the use of too high-priced special types of steel, to whatever extent that correcting the design will serve the end.

ENTITLED as they are to the greatest possible credit for their excellent work in motor car construction, which laid the foundation for our almost perfect vehicles of to-day, it must be admitted that, during the past two or three years, the greatest value of our automobile engineers has been their efforts

toward the standardization of materials and the parts used in motor cars. I am convinced there is the greatest possible kind of field for this standardization of motor vehicle construction, and the men are just as important a part of the automobile industry as the engineers are in other industries.

One of the greatest works in the future of automobile engineers is the designing of automobiles so as to make the greatest possible use of commercial materials, instead of so designing that fancy or peculiarly alloyed steels are required, with an increased cost to the public.

While there must always be a very large margin of safety, I refer particularly to the practice of backing up faulty designs by using expensive materials which involve an added cost.

Engineers must rise above themselves to the extent of admitting their errors and re-designing faulty parts and connecting parts instead of endeavoring to overcome errors by including materials unnecessary under ordinary conditions.

In a properly designed car, certain materials are needed and any materials costing in excess of those materials becomes a burden on the public. I find too often a tendency on the part of engineers to back their own ideas to the extent of using materials that involve a 100 per cent increase in the cost to gain little or no efficiency.

Standardization in the future will be marked by more pronounced features than in the past and I expect to see so important and creditable an organization as the Society of Automobile Engineers take the leadership in the work.

By standardization, I mean among other things such work that the man owning one of your cars will not have it disabled waiting to secure a part from the factory, especially when the part is of minor importance. We feel that our duty to the public requires us to make our cars as simple and as near a standardized form as possible.

In the near future I hope that our engineers and our manufacturing departments can produce cars for the United States

Motor Company so that the owner need not lay up his car for a number of days waiting for a nut or a bolt to be shipped from the factory, because that nut or bolt happens to be of a certain thread. I hope to see a standardized thread so that the owner of a car can secure a nut or bolt from the average hardware store (even if it is not of as good material) that will operate until the proper part can be shipped from dealer or branch.

Settling down to one point, it therefore seems that the greatest field for the automobile engineers is standardization and the calibre of the men in our industry makes me feel sure that they will rise to the occasion in a manner that will assure our continued supremacy in the motor car industry.

## Province of the Engineer

*C. R. Culver, Sales Manager of the Knox Automobile Company, considers that upon the shoulders of the engineer rests the reputation of the product of a factory, and outlines some of the duties that devolve upon him if the cars that he designs are to be up to the expectations of the buying public.*

THE engineering department is one of the most important in the up-to-date automobile factory. Upon the shoulders of the head engineer rests the reputation of the product, for unless he is capable of designing a car that is graceful in body, powerful in motor and able to stand abuse without breakdowns, his product is a failure from the market standpoint.

The engineer must be a man of experience and one who keeps constantly posted on the experimental results of others, for otherwise he may waste months of time and much money trying out ideas which have already been found impractical by others.

He must also drive his models, from the racing car to the six-ton truck, as much as his time permits, so that he can see for himself the mechanical troubles which may arise in the hands of customers and eliminate them before the model gets to the public.

While keeping well up on the mechanical end, he should not forget the object for which the cars are made—to sell, and the fact that what the public wants one season is not necessarily what it may want the next year. Toward this end he should co-operate with the sales department in forecasting the demand and building what the public is looking for.

It is necessary, by all means, that engineers should get together, exchange ideas, talk over mistakes, forecast conditions and note the general trend of the motor industry. This is an immense help to the engineer, the manufacturer and the buyer.

## Exacting Work Must Be Done in the Laboratory

*John N. Willys, president of the Willys-Overland Company, makes a timely statement of the functions and duties of the engineer in the plant and tells of the things that are being done to make automobiles reflect engineering attention. It is suggested also that there are certain limitations that must be kept in mind in dealing with the production side of the automobile art.*

THE possibilities of the future for the engineering department of the automobile industry range almost beyond thought.

Until quite recently practically all methods have been empiric, and many of the failures of the past have been due not so much to lack of business methods as to freakishness of design and impracticability.

The last few years have shown the necessity for conservatism in automobile design, and the desirability of moving along the lines of least resistance. The practice of that conservatism has evolved what is known as a standard design.

Experimentation has, however, by no means ceased—nor will it—but the business men in the automobile industry of to-day

are less prone to adopt anything that seems new, unless its practicability can be undeniably demonstrated.

This should not prevent engineers from working out their individual ideas; it will merely have the effect of making them sure of their success before the ideas can be adopted.

The interchange of ideas among engineers should surely work for benefit of all of them, and for that reason I am glad to see these meetings promulgated.

The old-time engineer who could design a motor so that it would run, has lost whatever usefulness he may have possessed, and the present day engineer must possess a knowledge of the sciences, metallurgy and chemistry, in order to become a success. Hit or miss methods no longer prevail, because they are not marketable. Exact analysis of material is an absolute essential, and no manufacturer can risk his reputation upon a car that combines other than the material that chemical and metallurgical analyses have shown to be the best for its purposes. The automobile industry, for instance, has made vanadium known, and I see no reason why further discoveries of the metals in the future, by the interchange of experimental knowledge and of the results of chemical experiments, should not add greatly to the knowledge we already possess.

An engineer must necessarily be somewhat imaginative to be inventive—he is largely a speculation wherein others pay his losses—and it is well to remember that without the hard-headed, practical business man, while the engineer would be an interesting element in the industry, he would scarcely be productive from a financial standpoint.

## Experience Counts in the Long Run

*F. H. Trego, chief engineer of the E. R. Thomas Motor Company, in giving his views upon the position that the automobile engineer holds in the plant, expresses the opinion that this important member of the automobile manufacturing corps holds entirely too much aloof from intimate contact with the dealers in automobiles and the purchasers thereof.*

AUTOMOBILE engineers do not pay enough attention to the Service Department of his factory, nor to the workmen throughout the factory.

In order to obtain the best results the automobile engineer should frequently come in contact with the dealers and encourage them to make suggestions and criticisms. He should constantly convey the impression that his office is not a throne room, but simply a work-room like the blacksmith shop, and encourage the men throughout the shop to come to him with any suggestions they may have, and make them feel entirely at home, so that they will put forth their argument for the suggestions, and take his argument against it or for it, in good part. I have obtained invaluable suggestions many times from blacksmiths, assemblymen, machinists, etc., and engineers in general will find that they cannot afford to "ride a high horse" if they are to produce the best results.

An engineer should drive a car constantly, so that he will be convinced by practical demonstration of the faults of the car.

Gathering all of the suggestions of the agents and others, the engineer should combine these, together with his own ideas, into the design of the car. Seventy-five per cent. (75%) of the car should be evolved from practical experience and the ideas of practical men. The other twenty-five per cent. (25%) might be theoretical ideas which the engineer could experiment with in the experimental room, and from the practical experience thereby gained, determine whether or not these ideas should be incorporated into the design.

A very good motto for an engineer is "Stop, look and listen," the last word applying most particularly to the murmurs arising from the repair department, and the ideas and suggestions of the dealers.

## Hugh Chalmers Says: "Automobile Engineers Made Industry"

*According to Hugh Chalmers, President of the Chalmers Motor Company, Detroit, Mich., the industry owes its inception to the engineer type of man, and the stamina of the industry is entirely due to the conservative activities of the mechanical man. Praise for the co-operative trend of the Society's activities and the example it has set for the industry in establishing standards are acknowledged by Mr. Chalmers, who happily concludes his address with a declaration of loyalty and a ringing call to motordom to support the organization.*

THAT the fate of the automobile industry rests in the hands of the men who created it is fitting. That the wonderful success of the automobile art is to be directly traced to the automobile engineer is plain. That the imperfections, if such there are, belong to the engineer to fix, may also be taken for granted, and that the great achievements that may be seen in the form of progressive automobile manufacturing plants at every hand are to be accredited to the genius and activity of the automobile engineer is also true.

In congratulating this splendid body of men upon the great results that have grown up under its care, and in direct line with its watchful eye, it must not be forgotten that this body is international in scope and that the brother engineer in Europe must be given his fair measure of recognition along with the rest.

In tracing the rise of the automobile industry from its position of a practical nonentity a mere decade ago to its place as the third industry in point of magnitude and worth in America at the present time, this marvelous situation must be ascribed in the main to the underlying worth of the plan from an economical point of view, and for its healthy growth to the conservatism of the automobile engineer.

In a concrete way, recounting the work that has been done, under the fostering care of the engineer, mention must be made of the fact that the high priced automobile of yesterday is the relatively low-priced car of to-day, and that quality as it obtained in the original conception without a price, parades before the purchaser of present product with a close limit on the cost to him. In other words, the automobile engineer, by his work and his example, has maintained, or better yet, advanced, quality, not without reducing cost.

But the automobile industry is still young. There is no precedent to its back upon which to lean, and the activities of these leading men must be without guidance. Something of this situation is reflected in the fact that until recently the various manufacturers of automobiles blazed their own path, working along individual lines.

The Society of Automobile Engineers is setting a good example by its co-operative trend, and in the establishing of standards to go by, with tables of measurements and other data, to aid in the designing and building of cars, it is showing its acumen.

The amount of good that is to come out of this standardization work is not to be estimated in the light of the present, but when we consider the large number of well made automobiles that are to be had at moderate cost, in view of the small amount of standardization work that has gone before, we are enabled to view the past with equanimity and enjoy the perspective that stretches out before us.

Let us not sink in the mire of pleasant generalities and think of the work that we have done as being of such magnitude and merit that we may now rest upon our well-earned laurels, nor forget the lessons learned. The day is past when any company can put a model upon the market, if all that can be said of it is to be confined to the beauty of its theory to the exclu-

sion of quality of material or excellence of workmanship, and of the serious things that must be found on the title page of the book of merit, the plan whereby the purchaser's investment may be conserved, should greet the eye.

Let us congratulate the Society of Automobile Engineers for the broad-minded attitude of its readers, and for the patient and loyal support of the body politic of this splendid organization, and join with it in the long up-hill haul that must lead from the excellence of our position of to-day to the better work that will be expected of us in view of our high aims, upon the part of those to whom we owe so much—our honorable purchasers.

## The Musings of a Day Dreamer

*Some more or less apropos expressions of current thought bearing upon the phases of the automobile industry, indicating in a measure the difficulties that arise when an idea is being staged for its practical application, indicating also how troublesome it is to work an improvement into the head behind the bank account, despite the excellence of its promise.*

ALL the world being a stage, there is just one very strange thing about it; namely, that the point of vantage of the spectator decides whether the play appears to him as a comedy or a tragedy, or, as in the most cases, and sad to relate, as a platitudinous melodrama in which he thinks that he plays a leading rôle. That his idea is a fallacy, is generally recognized only by the other spectators; and to bring about a truer aspect on his part is almost as hopeless a task as to move the planet from a fulcrum without.

In other words, the different impressions of the world of the various onlookers depend, to a considerable extent, upon their tempers; and the temper of a person is also a great factor in his struggle for success.

One may call it a tragedy or a comedy, but the fact remains that if the careers of a number of individuals are likened to a race, assuming an equal start for all participants, the outcome is in most cases very different from what even the wisest onlookers would have presupposed. Just as a good car may win in spite of the fact that its cylinders miss fire at the instant of starting, so may a sterling performer come to a brilliant finish after doing inconspicuous work at the outset. Yes, it may even be said that in the majority of any given number of races the winners went to the lead only during the last few laps, and that steady persevering handling of a machine proved of higher value in the long run than a brilliant performance during the early stages of the event. A rapid pace could not possibly be kept up for a continued length of time.

It is in accordance with this rule that they who shine brilliantly at the age of fifteen or twenty have exhausted their energies before the real battle begins; when they are out of school, their heads are indeed well filled with the records of the scientific achievements of the last dozen years, but it is the hardest thing conceivable for them to step forward independent of a guide, such as has led them up to that time.

They have made remarkable efforts in bringing into harmony the contradictory contents of scores of books; and after this having been achieved, they themselves have been turned into books—stores of knowledge which, however, require a new effort on the part of their fellows to be turned into practical value.

But this is not fully correct. These men are not mere books, because sometimes they give life to useful and even original ideas, if there is such a thing as an original idea. However, these ideas are developed, in general, in the company of men of the same class, and therefore remain thoughts. The more a proposed improvement of an existent practice is talked about

among some people, the less chance is there of its being put into practice; first, because much time and energy are wasted in the output of words, and because when a bullet's whistle is heard it has missed its chance of hitting; second, because it requires a special kind of effort to put an idea into practice.

"Within the head the thoughts are dwelling freely, but out in space are objects sadly crowded." The truth of this saying is unknown to all except the practical men, and when one of the other specie is confronted with the task of carrying into effect some golden thought, he is astonished to find that people have not been awaiting him and his project all the time, but rather resent its acceptance, and that it is left for him to work his discovery or plan of improvement into the head of the man behind the bank account. This fellow asks about cost and profit; and the unsophisticated and unexpecting reformer is sometimes so struck by an interruption he was not prepared for, that he gives up at once instead of fighting on.

Under the circumstances, he is hardly to blame if the causes for his behavior are understood. Nevertheless, it is disquieting to see a good man make a rank failure in a good cause. Of course, the looks of his adversary are disconcerting to him and must be so; but because money is kept behind bars, and criminals are, too, is there any reason in this coincidence for an honest man to be scared by the grim sight of iron bars? True, the man that the reformer is up against often looks anything but encouraging, still this fact does not excuse the man, who represents progress which must come, for backing out and retreating with a whine.

Furthermore, if there is an unpleasant tint about this situation, the blame must in at least as great a measure be laid upon the opponent of the reformer, not only because his way of acting hampers a general advance of the industry, but because of the unintelligence expressed by his stubbornness. The latter is of the greatest disadvantage to himself, so that he gets the worst of it if he thinks himself too experienced, too practical and too busy to listen to the proposals of the man who offers him an improvement in the shape of an idea.

For, undoubtedly, all improvements are first in this shape; and while it must be admitted that they are of no practical value in this form, i.e., as thoughts, the same can be, must be said of the money kept behind the bars.

Putting the situation in a nutshell and likening the automobile industry to an automobile motor, who will deny that all the fuel there is in the world will not make the engine go unless the ignition system functions properly; and on the other hand, that the best-made magneto in conjunction with a first-class sparking equipment will be of no use if it is not given an opportunity to produce its spark and a highly compressed and combustible mixture? Yet the industry is such a motor; the engineer reformer represents the ignition mechanism; and the other party the fuel which is the second indispensable factor in the workings and development of the industry.

## Report on Wheel Dimensions for Solid Tires

*Being the Report of the Division on Wheel Dimensions and Fastenings for Tires of the Standards Committee of the Society of Automobile Engineers as of date of May 11, 1911, and subject to the vote of the Standards Committee.*

AS the result of six meetings of your committee since its appointment in January, 1911, a conference with the representatives of tire and wheel manufacturers, and considerable supplementary correspondence with these tire and wheel manufacturers and with vehicle builders, your committee desires to make its final report on that division of its work relating to wheel dimensions for solid tires and therefore respectfully submits for your acceptance and approval its findings as expressed in the following resolutions passed at our recent meeting held in New York, May 4, 1911:

"Resolved, That the Wheel Dimensions and Fastenings for Tires Division of the Standards Committee recommend for acceptance by the Society of Automobile Engineers the following as standards for dimensions of wheels for solid tires:

"There shall be a permanent metal band on all wheels in advance of shipment by wheel manufacturers.

"This metal band shall be one-quarter (1-4") inch thick on wheels for single tire equipment up to and including four (4") inch nominal width of tire; and three-eighths (3-8") inch thick on wheels for single tire equipment above four (4") inch nominal width of tire, and three-eighths (3-8") inch thick on wheels for dual equipment of all tire sizes.

"WHEEL DIAMETERS—There shall be a constant wheel diameter over the metal band for all widths of solid tires of a given nominal overall tire diameter. Nominal tire diameters shall increase or decrease from thirty-six (36") inches by even two (2") inches.

"The constant diameter for a wheel taking a thirty-six (36") inch tire shall be thirty (30") inches over the metal band, and shall increase or decrease by even two (2") inches for tires larger or smaller than thirty-six (36") inches.

"WIDTH OF FELLOE AND WIDTH OF BAND (Single Tires)—The width of felloe and band for single tires shall be three-quarter (3-4") inch less than the nominal width of the tire equipment for same. For example, the standard felloe width for four-inch single tire equipment shall be three and one-quarter inches.

"DUAL TIRES—The width of felloe and band for dual tire equipment shall be twice the nominal width of each of the dual tires. For example, the standard felloe width for four (4") inch dual tire equipment shall be eight (8") inches.

"TOLERANCE OVER METAL BAND—The tolerance allowable in the circumference of metal bands shall be as follows:

Before application to wheel

Minus	0
Plus	1-16"

After application to wheel

Minus	0
Plus	1-8"

"MINIMUM DEPTH OF WOOD FELLOE—The minimum depth of wood felloes shall be as follows:

Tire Size.	Felloe Depth.	Tire Size.	Felloe Depth.
2"	1 1/4"	6"	2 1/4"
2 1/2"	1 1/2"	6 1/2"	2 3/4"
3"	1 3/4"	7"	2 3/4"
3 1/2"	1 3/4"	7 1/2"	2 3/4"
4"	1 3/4"	8"	2 3/4"
4 1/2"	1 3/4"		
5"	2"	Above 8"	2 3/4"
5 1/2"	2"		

FOR DEMOUNTABLE TIRE EQUIPMENT—All of the foregoing provisions for non-demountable tire equipment standards, viz.: equipment of wheels with metal bands, the constant diameter of wheel over metal band, the stated variation of wheel diameter, the width of felloe, the tolerance over metal band and minimum depth of wood felloe, shall apply to wheels for demountable tire equipment.

"DISPOSITION OF REPORT—Resolved, That this report be presented to the Standards Committee for its approval, with the recommendation that, in view of the urgent commercial necessity for decision in settlement of the standards in question, a mail vote of the Standards Committee be taken and rendered promptly to the Council of the Society for approval as to submission to the entire S. A. E. membership for final acceptance."

It is the opinion of your committee that the ground has been so thoroughly covered as to warrant definite and final action without further delay and that the commercial use of these standards should go into effect not later than January 1, 1912, as the intervening period appears to provide reasonable allowance for the manufacturing changes involved.

Respectfully submitted,

WILLIAM P. KENNEDY, Chmn.  
C. B. WHITTELEY,  
J. M. MACK,  
C. L. SCHWARZ,

E. R. WHITNEY,  
C. B. HAYES,  
J. A. ANGLADA,  
COKER F. CLARKSON, Secy.

## "Voice from the Promised Land"

Q And there came into the land a manufacturer, who did make with much cunning and skill great steel wagons that moved swiftly with many snortings and much odor.

Q And behold these wagons needed neither horses nor oxen, for with incredible speed they darted hither and thither, propelled by some unseen and powerful demon.

Q It came to pass, then, that the people did greatly desire the steel wagons and gathering much of the coin of the realm, they stood before the manufacturer who so cunningly devised them, saying:

Q "Make us now, we pray thee, wagons to thy liking, according to thy plans and with thy skill make thou them, for behold we stand before thee with many of the shekels of the ungodly and verily they do burn great holes in our pockets.

Q "Be merciful and take this uncomfortable wealth and deliver in lieu thereof, the demon wagons that we may scoot from place to place and make faces at the railroads."

Q And the manufacturer took pity upon the people and he promised them relief from their burning shekels, saying:

Q "Behold, I will make for thee wagons of perfection—I will build them according to the 'hunches' of my inventors and they shall be perfect in design, material and workmanship.

Q "Within their roomy bodies thou canst sit and ride—yea even millions of miles canst thou go with nary a breakdown.

Q "Verily I say unto thee, that even though thou climb and coast the dizzy heights of Mount McKinley with the wagons I shall make, or fall into the yawning Grand Canyon of Arizona, yet will not one bolt start nor the smallest part fail thee.

Q "Behold, I, the man of promises, swear to thee that thou canst not break steel wagons I shall build, even though thou 'swat' them with a fifty-ton steam hammer. Lo, I have spoken."

Q And the people were glad and rejoiced with exceeding joy as

they spoke of the great manufacturer. Verily they sang aloud and made genuflections before the prince of promises.

Q And it came to pass that the steel demon wagons were delivered to the people and they did climb merrily aboard, and with sundry "honk honks" and many roarings they were off.

Q See, now, does not Smith's "Forty" strike a small rock and sustain a dislocation of the steering knuckle.

Q Behold the pitiful condition in which Brown's roadster limps into the repair shop with an empty, leaky radiator and one cylinder in a "sling."

Q Look upon the plight of Jones as his car stands silent on a deserted country road, with a burned-out bearing on the port side and a spring with a compound fracture on the star-board side.

Q And it came to pass that in the land a spirit of discontent arose and the manufacturer heard a murmur of portentous moment—and his heart told him that all was not well with those who had come forth with the coin of Caesar begging for vehicles which required neither horses nor oxen.

Q And Smith, Jones and Brown, et al, came and stood before the manufacturer saying:

Q "Didst thou not promise us wagons that could safely be thrown into the Grand Canyon without even starting a nut—

Q "And didst thou not assure us that we could coast from the dizzy peaks of Mount McKinley without breaking a screw or burning a brake lining—

Q "Behold now, thou hast not made good, for thy steel demon wagons have laid down on the smooth routes of the middle west—and they have 'busted' something fierce on the town streets of New England. Verily we are undone.

Q "Tell us, we pray thee, what is the probable distance between promise and performance and where can we buy an odometer to measure it?"

## Chronology of the Dayton Meeting

### How the S.A.E. Proposes to Spend Its Time

FOR the convenience of those who attend the Summer meeting of the S. A. E. at Dayton, commencing to-day, as well as for ready reference, the full program of the society meeting is here given with reference to division of time.

Thursday, June 15, 8:30 a. m. Opening address by the president, Henry Souther. Business Matters.—Reports of tellers of election of members. Treasurer's report. Announcement of membership vote on constitutional amendments. Professional Matters.—Reports of standards committee divisions. a. Iron and steel division. Henry Souther. b. Aluminum and copper alloys division. W. H. Barr. c. Seamless steel tubes division. H. W. Alden. d. Nomenclature division. P. M. Heldt. The Question of Long vs. Short-Stroke Gasoline Motors. Paper by J. B. Entz. Long Addendum Gears. Paper by E. W. Weaver.

Thursday, 2:30 p. m. Aeroplane flights at Wright Brothers' grounds. Baseball game at Fairview Park. Band concert at National Military Home. Inspection of factories. Automobiles will transport members and guests from hotels to the various places desired.

Thursday, 8 p. m. Professional Session.—Commercial Vehicles. The Influence of the Engineer on the Sales Department. Paper by William P. Kennedy. Report of wheel dimensions and fastenings for tires division. Topics for Discussion.—Special methods of loading commercial vehicles. Dumping trucks. Auxiliary apparatus for commercial vehicles. Trailers for commercial vehicles. Location of working and emergency brakes.

Friday, 8:30 a. m. Professional Session.—Elements of Ball and Roller Bearing Design. Paper by Arnold C.

Koenig. Worm Gear and Wheels. Paper by E. R. Whitney. Reports of standards committee divisions. e. Ball bearings division. David Ferguson; f. Broaches division. Charles E. Davis; g. Carburetor division. G. G. Behn; h. Frames division. James H. Foster. Topics for Discussion.—Transmission location—whether on rear axle or attached to car frame. Underslung frames.

Friday, 2:30 p. m. Aeroplane flights at Wright Brothers' grounds. Baseball game at Fairview Park. Band concert at National Military Home. Inspection of factories. Automobiles will transport members and guests from hotels to the various places desired.

Friday, 8 p. m. Members will visit a theatrical performance at the Casino in a body.

Saturday, 8:30 a. m. Professional Session.—Rotary Valve Gasoline Motors. Paper by C. E. Mead. Some Points on the Design of Aluminum Castings. Paper by H. W. Gillett. Oversize Standards for Pistons and Rings. Paper by James N. Heald. Reports of standards committee divisions.—i. Lock washer division. J. E. Wilson; j. Sheet metals divisions. James H. Foster; k. Springs division. A. C. Bergmann; l. Miscellaneous division. Topics of Discussion.—Multiple-disc clutches. Six-cylinder vs. four-cylinder motors of equal rating.

Saturday, 1 p. m. Banquet at the Automobile Country Club, Hills and Dales. (No charge will be made to those attending the banquet.) Address by Arthur Ludlow Clayden, editor *The Automobile Engineer*, London, England.

Additional subjects for any session if convenient: Three-point vs. four-point suspension; current practice in lubrication and practical results obtained; elimination of noise in motor cars; present trend in compression of gasoline automobile motors; contests and engineering lessons which they teach.

## American Trucks for Jamaica Banana Planters

*There is a promising field in this British West Indian island for the freight automobile. Especially in the banana-raising industry does the field appear to be ripe. The many miles of excellent roads on the island offer excellent opportunities to American makers to show the ability of their product to perform the work.*

ONE of the firms conducting a banana plantation in the parish of St. Mary, Jamaica, has been testing an American-made, 50-horsepower, three-ton capacity motor truck for use on the premises. It is said that the machine is coming up to every recommendation that the manufacturer gave it. In fact, so well pleased is the planter that he has sent his agent to the United States to order two additional motor trucks for service.

Some of the banana planters have tried to organize a mutual liability company to operate a number of motor trucks. But not all of the planters have yet endorsed the plan. One of the arguments against it is that they lost money once before in a similar enterprise and they do not feel disposed to take stock in another motor car enterprise, until the utility and reliability of the cars now being tested for hauling bananas shall have been proven beyond the shadow of a doubt.

One of the motor cars in question which failed to come up to the mark was an English-built machine, which was shipped to

St. Mary's parish in Jamaica in September, 1910. The trial was to last six months. At last reports the car had traveled 1,200 miles; but no decision had then been given as to whether it would prove economical and practical over oxen and mule-drawn wagons. Early in July of the same year a planter had tested a 5-ton freight motor car for hauling bananas. It had proven a failure. A passenger and mail car trial between Kingston and Port Antonio and a motor car trial for carrying bananas in St. Thomas parish resulted unfavorably. One of the worst features of these tests seems to be the inability of the tires to grip the up-and-down grades in rainy weather, the soil being of a peculiarly slippery nature.

**COMPRESSION SHOULD VARY WITH THE SPEED**—If increasing compression demands increasing skill to match, as it does, beyond a certain point, the compression should be a maximum in racing cars, somewhat less for chauffeurs, a good mean for owners of pleasure cars and the minimum for taxicabs and commercial vehicles. This grading is on the assumption that it is good practice to sacrifice power to simplicity in proportion as simplicity takes on a commercial value. When a motor is rotating at a high rate of speed, as will be possible if the car is going on a hard level road, the effect of leakage of compression is minimized and losses of heat to the water-jacket will be less. As a result the compression pressure will be a maximum and it follows that the combustion pressure will be a maximum also.

## Calendar of Coming Events

### Handy List of Future Competitive Fixtures

#### Race Meets, Runs, Hill-Climbs, Etc.

June 15-16.....	Chicago, Ill., Reliability Run, Chicago Automobile Club.
June 15, 16, 17.....	Dayton, O., Midsummer Meeting Society of Automobile Engineers.
June 16-17.....	Milwaukee, Wis., Track Races, Milwaukee Automobile Dealers.
June 17.....	Philadelphia, Sociability Run for Electrics, Quaker City Motor Club.
June 17.....	Guttenberg, N. J., Track Races.
June 17.....	Portland, Me., Hill Climb, Maine Automobile Association.
June 18.....	Kenosha, Wis., Track Races.
June 19.....	Des Moines, Iowa, Annual Tour, Hyperion Field and Motor Club.
June 20-23.....	Detroit, Mich., Summer Meeting National Gas and Gasoline Engine Trades Association.
June 24.....	St. Louis, Mo., Reliability Run, Auto Club of St. Louis.
June 30.....	St. Louis, Mo., Reliability Run, St. Louis Automobile Manufacturers' and Dealers' Assn.
June .....	Denver, Col., Reliability Run, Denver Motor Club.
June .....	Norristown, Pa., Hill Climb, Norristown Auto Club.
June .....	Oklahoma, Reliability Run, Oklahoma Auto Association.
July 1.....	Ossining, N. Y., Hill Climb, Upper Westchester Auto Club.
July 1.....	Baltimore, Md., Hill Climb, Automobile Club of Maryland.
July 3-4.....	Brighton Beach, N. Y., Track Races.
July 4.....	Kansas City, Mo., Track Races, Automobile Club of Kansas City.
July 4.....	St. Louis, Mo., Reliability Run, Missouri Automobile Association.
July 4.....	Bakersfield, Cal., Road Race, Kern County Merchants' Association.
July 4.....	Denver, Col., Track Races, Denver Motor Club.
July 4.....	Detroit, Annual Track Meet, Wolverine Automobile Club.
July 4.....	Pottsville, Pa., Track Races, Schuylkill County Centennial.
July 5-22.....	Winnipeg, Man., Fourth Canadian Competition for Agricultural Motors.
July 7.....	Taylor, Tex., Track Races, Taylor Auto Club.
July 8 or 15.....	Philadelphia, Track Races, Belmont Park, Norristown Auto Club.
July 14.....	Philadelphia, Commercial Reliability Run, Quaker City Motor Club.
July 14-17.....	Reliability Run, Minnesota State Automobile Association.
July 15.....	Worcester, Mass., Hill Climb, Worcester Automobile Club.
July 17-19.....	Cleveland, O., Three-Day Reliability Run of the Cleveland News.
July 17-22.....	Wisconsin Reliability Run, Wisconsin State Automobile Association.

July .....	Amarillo, Tex., Track Races, Panhandle Auto Trade Association.
Aug. 1.....	Chicago, Ill., Commercial Reliability Run, Chicago Evening American.
Aug. 3-5.....	Galveston, Tex., Beach Races, Galveston Automobile Club.
Aug. 12.....	Philadelphia, Reliability Run, Quaker City Motor Club.
Aug. 25-26.....	Elgin, Ill., Road Race, Chicago Motor Club.
Aug. ....	Denver, Col., Hill Climb, Denver Motor Club.
Sept. 1.....	Chicago, Ill., Commercial Reliability Run, Chicago Motor Club.
Sept. 1.....	Oklahoma, Reliability Run, Daily Oklahoman.
Sept. 2-4.....	Brighton Beach, N. Y., Track Races.
Sept. 2-4.....	Indianapolis Speedway, Track Races.
Sept. 4.....	Denver, Col., Track Races, Denver Motor Club.
Sept. 7-8.....	Philadelphia, Track Races, Philadelphia Auto Trade Association.
Sept. 7-9.....	Hamline, Minn., Track Races, Minnesota State Automobile Association.
Sept. 12-13.....	Grand Rapids, Mich., Track Races, Michigan State Auto Association.
Sept. 15.....	Knoxville, Tenn., Track Races, Appalachian Exposition.
Sept. 16.....	Syracuse, N. Y., Track Races, Automobile Club and Dealers.
Sept. 23.....	Lowell, Mass., Road Race, Lowell Automobile Club.
Sept. ....	Denver, Col., Track Races, Denver Motor Club.
Oct. 3-7.....	Danbury, Conn., Track Races, Danbury Agricultural Society.
Oct. 7.....	Philadelphia, Fairmount Park Road Race, Quaker City Motor Club.
Oct. 9-13.....	Chicago, Ill., Thousand-Mile Reliability Run, Chicago Motor Club.
Oct. 16-18.....	Harrisburg, Pa., Reliability Run, Motor Club of Harrisburg.
Oct. ....	Atlanta, Ga., Track Races, Atlanta Automobile Association.
Nov. 1.....	Waco, Tex., Track Races, Waco Auto Club.
Nov. 2-4.....	Philadelphia, Reliability Run, Quaker City Motor Club.
Nov. 7-10.....	Los Angeles-Phoenix Road Race, Maricopa Auto Club.
Nov. 9-11.....	San Antonio, Tex., Track Races, San Antonio Auto Club.
Nov. 10.....	Phoenix, Ariz., Track Races, Maricopa Automobile Club.
Nov. 30.....	Los Angeles, Cal., Track Races, Motordrome.
Nov. ....	Savannah, Ga., Road Race, Savannah Automobile Club.
Dec. 25-26.....	Los Angeles, Cal., Track Races, Motordrome.

#### Foreign Fixtures

June 24.....	Boulogne, France, Coupe des Voiturées.
June 25.....	French Light Car Race, Coupe des Voitures Légères, Boulogne-sur-Mer course.
June 25-July 2.....	International Reliability Tour, Danish Automobile Club.
July 4 (to 19).....	Start of the Prince Henry Tour from Hamburg, Germany.

# THE AUTOMOBILE

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No. 24

## THE CLASS JOURNAL COMPANY

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GETTING all there is in an automobile is the problem of the user thereof, but putting stamina into the structure is the function of the engineer, if the head of the company will consent to the workings of the enterprise. Some great mind, depressed by circumstances, perhaps, found relief in the utterance, "Hell is paved with good intentions." No serious-minded person can bring himself to the belief that a real engineer harbors anything but good intentions, and yet when the truth is told every failure in the shape of an automobile of which there is a record is, at least in some part, the handiwork of an engineer. Of these failures, were the details classified, it would show that a poor understanding of the conditions was at the bottom of some of them, that lack of foresight made its mark in a few instances, but that the handicap of commercialism stood in the way in the majority of cases. At the seat of government of the Empire State there is an agitation going on among the senators, the idea being to put a law upon the statute books that will make it illegal for any newspaper to present an editorial of any character whatsoever unless the author's name is signed thereto. It is not the purpose here to discuss the merit of the signed editorial proposition, but there is a parallel thought which takes expression in the suggestion that the number of mistakes that get into automobiles would be very materially reduced were the designers thereof required to put their signatures upon the finished product. No real engineer would ever put his O.K. on

a finished automobile if he knew that the material as specified by him originally was thrust to one side.

\* \* \*

LIVING in an age of publicity, under conditions in which the truth may be disregarded in advertising statements, remembering also that truth is buried in glittering generalities in a large part of the editorial utterances that greet the eye of the purchasing public, not forgetting that the seller's agent has a loud voice and a confident air, it devolves upon the body politic of the automobile engineers as they assemble at Dayton, Ohio, to assert themselves, and if they possess the skill that is accredited to them they have a perfect right to let their power be felt and to serve an injunction on the rest of the world, one, in fine, that will bear evidence of their stamina, so that in future their light will shine as a beacon illuminating the darkened seas of uncertainty for no worse purpose than to prevent men who would sell their souls for a dollar from saying that "the engineer" has passed upon the plan, and that it is a worthy project.

\* \* \*

IDEALISM has a small place in commercial activities, and yet a trace of this impracticable characteristic is a better paying investment than it is to cast ideals to the four winds and to do business on the basis that a good salesman can sell anything. It is worth something to remember that a good salesman may be regarded in the light of a costly luxury, or he may take on a profitable phase, depending merely upon the character of the duty that is set for him to perform. Referring to automobiles, if a good salesman is charged with the task of disposing of praiseworthy product he will make rapid inroads upon the available supply, and he may then be looked upon in the light of a profitable investment. But if the good salesman must also be a conniving miscreant, if he must call defects by another name, and if he must "hypnotize" the unwary, there will be a "string" attached to every transaction that he consummates, and there will be a loss to sustain on the part of the exploiter in the long run, since he, too, will be the victim of his mal-enterprise.

\* \* \*

NOW that racing and contests generally have been reduced to a low ebb, due to the chicanery of "sports," the wonder is that the makers of automobiles who know how thoroughly reliable their own engineers are would thrust them aside and place the responsibility for fair contest work in the hands of those who merely wish to get more money out of it than they can possibly be worth on any count whatsoever. It is probably a great mistake to allow politics to creep into a plain situation, and it is more than likely that the useless timber will have to be dumped and that racing and contests will have to be conducted under the guidance of those who have proven that they have not only skill but fairness; but this will be impossible unless the hampering influences of the mere parasite can be done away with. It is really laughable to have a technical examination of automobiles taken out of the hands of the men who have proven their worth by building them and put into the hands of men who, if they are struggling with the first principles of automobile designing, it is the best that can be said for them.

## Butler on the Future of Contests

### Chairman of Contest Board Outlines Views

*Chairman S. M. Butler, head of the Contest Board, gives his conclusions with regard to the questionable practices that have crept into the matter of advertising based upon the performances of automobiles in competition. He says that publication of results is more important to manufacturers than the performance of their cars, because the printed articles reach a vastly larger audience than could possibly witness a contest. Therefore he says the maker should be held responsible for the publication of reports by his agents.*

THE question, "What is the future of automobile contests?" may be answered by a consideration of the evolution of such contests up to the present time. Ten years or more ago, the first automobile contests were participated in and supported largely by amateur owners and drivers. The tremendously rapid growth of the automobile industry and the entry into the field of so many new makers, the increased speed of the cars, the expansion both in number and magnitude of contests of all forms, necessitating large outlays of money by the individual participants, soon caused a withdrawal of the participation of the amateur owner, and the consequent entry of the professional driver into the contest field.

Automobile contests at the present time must therefore be looked at from the viewpoints of the general public and of the manufacturer. The love of competition is innate in the human mind; the spectacle of the annihilation of space by the most modern method of locomotion—the highest attainable motor speed—is universally attractive to every one living in this Twentieth Century. To the manufacturer who participates in automobile contests they offer an opportunity for the public exhibition of his product and the detection of mechanical defects as a result of long continued high speed, which he may correct in the car he sells to the public. The public is therefore the direct beneficiary of such contests, for the correction of the defects so detected provide a large margin of safety in the stock car of such maker in the hands of the individual under ordinary conditions of use. Similar advantages accrue to the manufacturer and the public from the holding of touring contests wherein the public may note the actual performance of the regular stock car offered for sale by the manufacturer.

It is axiomatic that automobile contests, to hold the public interest, must provide keen and honest competition; it is equally true that such conditions cannot be maintained without the rigid enforcement of rules and regulations by competent and honest officials and the absolute and unhesitating co-operation and willingness to comply with the rules on the part of all participants in such contests.

The future of automobile contests depends largely on keeping faith with the public. The medium through which the public is reached is largely printers' ink. Five thousand people may witness a contest; five hundred thousand will read about it in the news and advertising columns of the daily and trade press. It is vital that the rules be enforced during the running of a contest; it is vastly more vital that the advertisements of the performances of cars in such contest, after it is run, be strictly in accordance with the facts in order that the thousands who are unable to witness the contest, and who get their only information

through the press, should not be misled by adroitly worded advertisements which leave their readers floundering in a sea of doubt as to which car or what character of car was the real winner. To attempt to figure a "printers' ink" or "paper" victory out of an actual defeat, or to say "We would have won if certain things had not happened" is suggestive of the old fable of the tortoise and the hare and does not make for any convection in the public mind as to the bona fides or honesty of automobile contests or the published accounts thereof. Rigid rules for the prevention of such misstatements are laid down by the Contest Board. Upon the degree with which such rules are observed depends, in no little measure, the successful future of automobile contests.

One and only one practical method exists of providing the means whereby the public may be protected from misleading advertisements of performances, and it is this: To furnish a copy of the full official report of every contest (which under the rules is required to be immediately transmitted to the Contest Board) to every entrant in a contest, and to hold such entrant and his agents and representatives to a strict accountability that his or their advertising conform to such report.

### N. A. A. M. to Hold New York Show

When January rolls around again and 1912 is well settled in its course, the annual show season will be the feature of the day. Next Winter there will be two shows in the Metropolis as usual, not reckoning in the Brooklyn exhibition and that held at Newark. The exhibition in Madison Square Garden will be given by the Automobile Board of Trade as the duly accredited successor of the A. L. A. M. and in addition the N. A. A. M. has arranged to give a show with coincident dates.

The directorate of the N. A. A. M. held a meeting this week at which plans were discussed and tentative arrangements perfected. Among those present at the meeting were William E. Metzger, S. D. Waldon, Thomas Henderson, Charles Clifton, Alfred Reeves, R. D. Chapin, L. H. Kittredge, W. T. White, A. L. Pope, J. W. Gilson and S. A. Miles.

The idea contemplates the accommodation of all manufacturers who are not members of the Board of Trade, in conjunction with the members of the N. A. A. M., who are without the membership roll of the Madison Square Garden exhibitors.

Naturally enough this includes the concerns that exhibited at the Grand Central Palace last winter, whether they are members of the N. A. A. M. or not. Half a dozen of these factories have been formally reinstated by the National Association and it is announced that all the others are welcome to reinstatement to eligibility.

Search is now being made for adequate quarters for the show and it is stated that three available buildings are under consideration. It is likely that one will be selected somewhere near the Garden, but definite action will not be taken in this regard until the regular meeting scheduled for July 7, at least.

A working agreement has been arranged between the N. A. A. M. and the Motor and Accessory Manufacturers so that in future the traffic matters of the M. A. M. will be handled by the Traffic Department of the N. A. A. M. This arrangement commences July 1 when Traffic Manager James S. Marvin is making preparations to take charge of this vast additional amount of work.

## Record Crowd, Good Fields and



A—Crowds and cars returning home after the races at the point where the hill begins to steepen

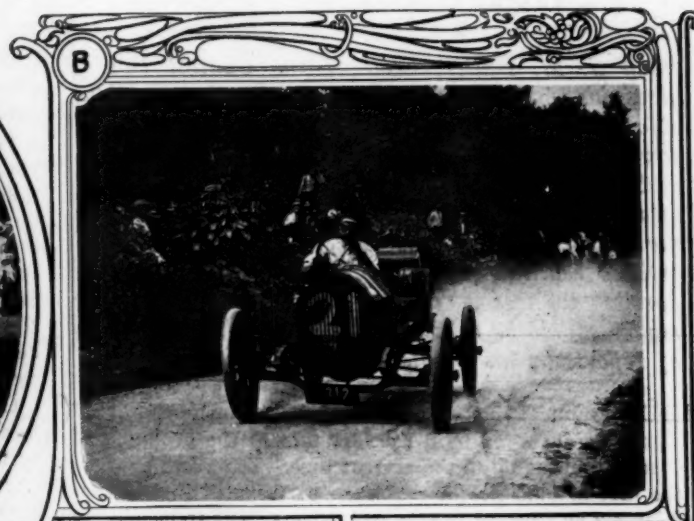
With well-balanced fields in each event, the annual hill-climb of the Yale and New Haven Automobile Clubs on Saturday proved to be an interesting contest from first to last. The races were sharply fought out and some desperate driving resulted in each class. The free-for-all was won by Bruce-Brown in a special Fiat in time that marks a new record for the hill.

**B**EFORE the greatest crowd that ever gathered to witness a hill-climbing contest in New England, the annual climb of the Yale and New Haven Automobile Clubs was held Saturday on Shingle Hill at West Haven. It was a thoroughly satisfactory event in which each class attracted a good field and developed a sharp contest. The stock car program which had been advertised was abandoned because of the small number of cars that would have been eligible to go and the consequent reduction of the fields in these events, but two of the climbs were fought for by simon-pure stock cars.

The honors of the day went to a six-cylinder Knox of 60-horsepower, which won its class event in 52.75, defeating the Fiat car which was driven by Bruce-Brown in the recent Indianapolis 500-mile race. In the field also were a Palmer-Singer and a Houpt-Rockwell. The Knox made the climb with a margin of over three seconds to spare.

In the class events, the Empire 17 won the race for little cars, closely pressed by another Empire. The Hupmobile was a rather distant third. The next higher class went to the Ford 25, in a tight fit, Paige-Detroit was second, a little over a second behind, and Buick 23 was a close third.

S.P.O. 31, Correja 68 and Metallurgique 30 were the placed cars in the 231-300 cubic-inch piston-displacement class, while



B—Paige Detroit coming up the hill in good style. Second in event 5



C—Empire, winner of event 4 entering the S turn

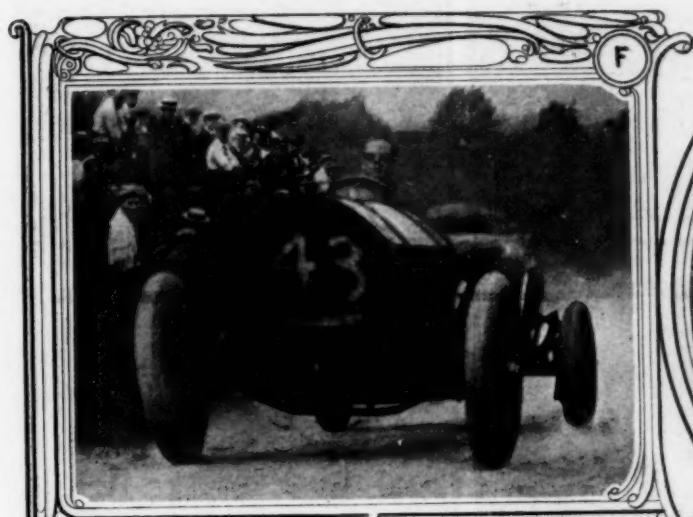
D—National in Class C after passing the S bend, where a large proportion of the spectators flocked



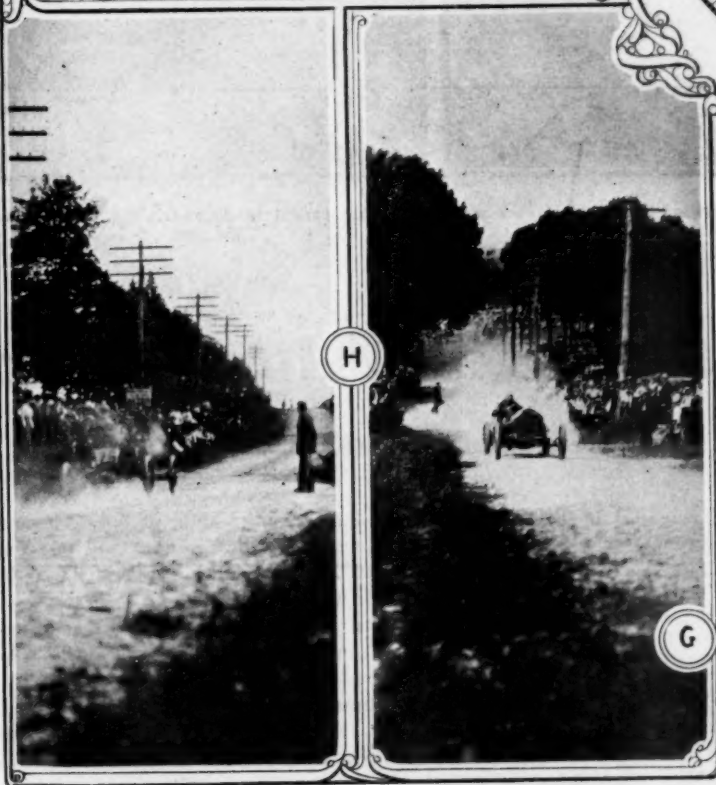
Marion 35, contesting with an amateur driver, would have taken second place had the car been eligible to compete with the professionals.

In the 301-450 class, National 38, driven by Caleb Bragg, won handily from a four-cylinder Knox and an Amplex. The 600-cubic-inch class went to the big Knox, as has been told, and in the amateur section of this event, Fiat 49 took the cup in 1:00.15 from a Simplex and two Stearns cars.

# Fast Time at Shingle Hill Climb



E—View of the S turn between the two hills, where some spectacular driving was witnessed



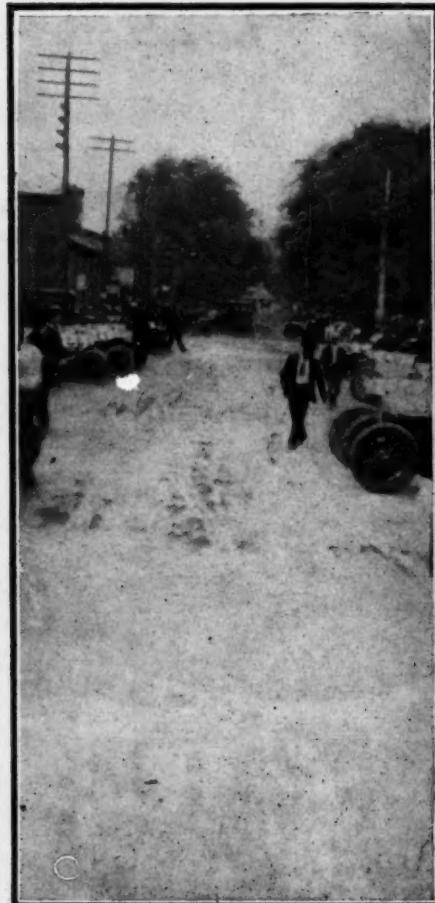
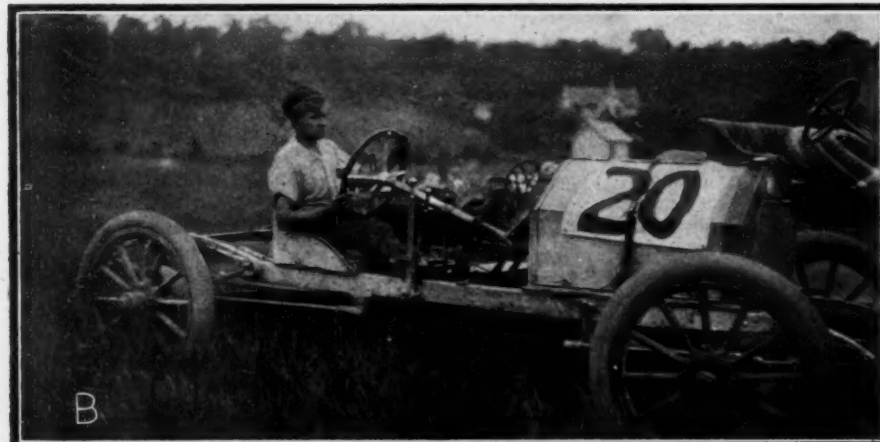
F—The Knox car, No. 43, which landed the 451-600 class, and came in an easy second in free-for-all

G—Stearns making good headway near the top of the hill

H—Looking up the first hill with the Ford, No. 25, winner of event 5

The free-for-all developed some of the most spectacular driving ever seen on a hill. The winner, a 200-horsepower Fiat, driven by Bruce-Brown, broke the record of the hill by six seconds, taking more chances than were comfortable, but meeting with no mishaps. The big Knox was second, making about the former record, and the Indianapolis Fiat was third under Matson's handling. In the amateur section, Simplex 70, driven by Heitmeyer, was a handy first, with Rutherford's National second and Simplex 50 third. Summary:

160 Inches and Less, Class C				
No.	Car.	Driver.	Position.	Time.
17	Empire	Hotchkiss	1	1:21.52
14	Empire	Kaerer	2	1:22.22
16	Hupmobile	Bishop	3	1:46.62
161-230 Inches, Class C.				
25	Ford	Smith	1	1:13.25
21	Paige-Detroit	Craig	2	1:14.55
23	Buick	Bull	3	1:14.82
20	Oakland	Bauer	4	1:18.36
24	Cutting	Lee	5	1:27.35
22	Regal	Stevens	6	1:29.58
For Amateur Drivers				
47	Buick	Hooker	1	1:24.05
231-300 Inches, Class C.				
31	S. P. O.	Robinson	1	1:08.09
68	Correja	Brainerd	2	1:13.17
30	Metallurgique	Bragg	3	1:14.35
32	Cutting	Strauss	4	1:16.55
33	Marion	Gillam	5	1:18.82
29	Correja	Lee	6	1:21.97
Amateur Drivers				
35	Marion	P. G. Thebaud, Jr	1	1:12.57
301-450 Inches, Class C.				
38	National	Bragg	1	54.27
36	Knox	Coffey	2	56.50
39	Amplex	Jones	3	58.85
37	Oakland	Bauer	4	1:20.05
41	Buick	Bull	5	1:20.82
451-600 Inches, Class C.				
43	Knox 6-cyl.	Belcher	1	52.75
28	Fiat	Matson	2	55.98
64	Palmer-Singer	Schoeneck	3	59.70
44	Haupt-Rockwell	Cottier	4	1:03.35
Amateur Drivers				
	Steinbrugge		1	1:00.15
	Haas		2	1:03.77
	Tours		3	1:08.11
	Iselin		4	1:14.04
Free-for-All Class D.				
	Bruce-Brown		1	45.29
	Belcher		2	51.74
	Matson		3	53.85
	Coote		4	54.37
	Bragg		5	55.48
	Coffey		6	56.16
Amateur Drivers				
	Heitmeyer		1	55.34
	Rutherford		2	57.71
	Haas		3	1:02.20
	Iselin		4	1:12.50



A—Hearne in Benz that won in free-for-all

B—Staver-Chicago, driven by Monckmeier

C—Cars parked in Algonquin's streets before the race

## Duplex Climb Staged at Algonquin

### Program Divided in Halves for Two Courses

*Sixth renewal of the annual hill climb of the Chicago Motor Club developed many interesting contests in the various stock-car classes and the feature event, a free-for-all, went to a Benz racing machine by the narrow margin of one second, a Ford, being second in the big struggle, making much faster time in the second half than the winner. Half of the contest was run in the morning over the artificial hill made for the club and the other half was over a natural grade. The time made in the morning and afternoon was added in each case and the lowest figures took the cup. Aside from sheer speed, a formula was resorted to in determining winners in the touring car classes.*

**C**HICAGO, June 12—With well-ordered smoothness the annual Algonquin hill climb of the Chicago Motor Club was run off Thursday for the sixth time and a special Benz racing machine won the free-for-all feature event by one second margin from a Ford car and a National. There were numerous contests in Classes A, B, C and D, covering fully equipped cars, stripped stock chassis, piston displacement without weight limitation and free-for-all cars, respectively.

The arrangement was a duplex contrivance, including a climb of the artificial hill at Algonquin in the morning and one over Phillip's hill in the afternoon. The two times were added together and the car in each class which made the lowest time was declared the winner. Algonquin hill is 1,000 feet long with a maximum grade of 26 per cent. and one bad turn. Phillip's hill is half a mile long with a grade of 12 per cent. in spots.

About 2,000 persons witnessed the affair, which was run off without mishap of any sort. Walkovers were quite frequent during the day, but most of them were run off at speed and proved spectacular, even with the element of contest eliminated. The hills were in excellent condition, having been rolled and oiled and flying starts were made for both halves of the program.

One result of the double-barreled character of the contest was to keep up the uncertainty as to the winners until all the cars had finished, when a simple mathematical example showed who was out in front. For instance, the winning Benz car made the morning climb in 163.5 seconds and the Ford made it in 201.5 seconds. In the afternoon the Ford climbed Phillip's hill in 281.5 seconds, making its total 482.5 seconds. The Benz turned the second hill in 304.5 seconds, or 23.5 seconds slower than the Ford, but won on account of its fast showing in the morning.

This naturally had a tendency to keep down enthusiasm as many of the spectators were not as expert mathematicians as the



F—J. H. Seek in National, tied for second in his class



D—Kulick driving his Ford up the hill



E—Big field in the event for cars selling at \$800 and under

officials. The record of Phillip's hill is 28 seconds, made last year by a Chadwick 60-horsepower car, and, in order for the Ford to have won, it would have been necessary to break that record by about a second. Thus, in spite of its fast race over the natural hill, the stiff grade of the artificial hill imposed such a handicap upon the American car that it proved impossible to lift.

As appears in the summary attached, several of the events were closely contested in total elapsed time and the fields in a number of the races were excellent. A formula was used in determining the winners in some of the events, in which the fields were small.

The summary:

#### RESULT OF HILL CLIMB OF CHICAGO MOTOR CLUB, AT ALGONQUIN

Class A, Stock, Division 1 A, \$800 and Under				
No.	Car.	Driver.	A. M. Time.	P. M. Time.
2	Ford	Lanahan	:29 1-5	:51 2-5
1	Ford	Gruener	:30 3-5	:50 2-5
3	Ford	Kulick	:38 4-5	:49 3-5
Class A, Stock, Division 2 A, \$801 to \$1,200				
4	Oakland	Bauer	:29 1-5	:43 2-5
Class A, Stock, Division 3 A, \$1,201 to \$1,600				
7	Oakland	Bauer	:25	:43 2-5
Class A, Stock, Division 4 A, \$1,601 to \$2,000				
8	Velie	Cooney	:22 3-5	:37 3-5
9	Staver	Monckmeier	:26 3-5	:42 2-5
Class A, Stock, Division 5 A, \$2,001 to \$3,000				
10	National	Seek	:19 4-5	:35
Class B, Stock, Division 1 B, 150 Inches and Under				
6	Empire	Meddock	:30 1-5	:42 3-5
5	Empire	Anderson	:27 4-5	:53
Class B, Stock, Division 2 B, 161 to 230 Inches				
14	Abbott-Detroit	M. Roberts	:23	:36 4-5
12	Abbott-Detroit	M. Basle	:28	
Class B, Stock, Division 4 B, 301 to 450 Inches				
24	National	Seek	:17 2-5	:31 1-5
22	Velie	Stickney	:18 2-5	:33 1-5
23	Velie	Cooney	:20	:34

Class B, Stock, Division 3 B, 231 to 300 Inches				
20	Staver-Chicago	Monckmeier	:19 2-5	:33 4-5
16	Cole	Jenkins	:21 1-5	:35 2-5
18	Falcar	Gelnaw	:21 4-5	:35 3-5
21	Staver	Robillard	:21	:36 2-5
17	Falcar	Morris	:22 2-5	:37
19	Moon	Heinemann	:23	:37 3-5
Class B, Stock, Division 5 B, 451 to 600 Inches				
25	Falcar	Morris	:21	:36 3-5
26	Falcar	Gelnaw	:22 3-5	:36 1-5
Class E, Nonstock, Division 3 E, 300 Inches and Under				
37	Case	Jagersburger	:18	:32 1-5
28	Case	Strang	:19 1-5	:31 1-5
35	Staver-Chicago	Monckmeier	:19 4-5	:33 4-5
34	Staver-Chicago	Robillard	:20 1-5	:36 1-5
29	Falcar	Morris	:20	:36 3-5
30	Falcar	Gelnaw	:21 2-5	:35 3-5
31	Moon	Heinemann	:26	:38
36	Henry	Turgeon	:24 1-5	:41 4-5
33	Ohio	Mathews	:27 1-5	:44 3-5
Class E, Nonstock, Division 5 E, 600 Inches and Under				
38	National	Seek	:17 3-5	:30 4-5
Class D, Nonstock Free-for-All				
43	Benz	Hearne	:16 3-5	:30 4-5
46	Simplex	Soubiran	:17 4-5	:31 2-5
45	National	Seek	:18 1-5	:31 2-5
41	Velie	Stickney	:18	:33 1-5
42	Ford	Kulick	:24	:29 1-5
40	Velie	Cooney	:20	:33 4-5
Class C, Nonstock, Division 2 C, 161 to 230 Inches				
49	Ford	Gruener	:18	:32 4-5
48	Abbott-Detroit	Roberts	:20	:33 4-5
15	Velie	Stickney	:20 3-5	:33 4-5
47	Abbott-Detroit	M. Basle	:20 3-5	:38 3-5
50	Henry	Turgeon	:24	:40 3-5
Class C, Nonstock, Division 4 C, 231 to 300 Inches				
51	Case	Jagersburger	:17 2-5	:31 1-5
57	Case	Strang	:20 3-5	:31 3-5
52	Case	Jones	:20 3-5	:35
53	Falcar	Morris	:20	:38
55	Falcar	Gelnaw	:22 1-5	:36
54	Moon	Heinemann	:27 3-5	:37 1-5
56	Oh'o	Mathews	:27 4-5	:47 3-5
58	Cole	Jenkins	:55	:36
Class C, Nonstock, Division 4 C, 301 to 450 Inches				
64	Ford	Kulick	:20 1-5	:28 1-5
63	National	Seek	:17 3-5	:32 1-5
61	Velie	Stickney	:19	:33 2-5
60	Velie	Cooney	:19 4-5	:33 3-5

## Electrics Show Their Prettiest Qualities

*New York Dealers' Association conducts suburban run to show the service of the electric pleasure car for that type of transportation. The winners are to be decided according to closest approximation of a secret time that was fixed by officials of the Association. Both men and ladies took part in the contest as drivers, and luncheon was enjoyed by the whole party at the home of C. Y. Kenworthy at Bellerose, L. I.*

WITH fourteen contesting cars, representing three makes of electric pleasure vehicles, the suburban run of the Electric Dealers' Association of New York started from Fifty-ninth street and Central Park yesterday morning. The course was to Bellerose, L. I., the home of C. Y. Kenworthy, where luncheon was served, and return. The chief points touched were Elmhurst, Forest Hills, Jamaica, Little Neck, Douglaston, Bayside and Flushing. The total distance was 38 6-10 miles.

Prizes have been offered for the cars driven by the man and woman driver which come closest to a secret schedule time in which speed will be only one factor. This time will be determined by adding the estimates made by Mr. Kenworthy and Harvey Robinson and averaging the two.

In the contesting field were five Rauch & Langs, five Bakers and four Detroit electrics.

The officials of the tour were Mr. Kenworthy, of the Rauch & Lang company, Mr. Robinson of the Edison New York Company; Albert Weatherbee of the Detroit Electric Company and Nathaniel Platt of the Baker.

The purpose of the run was to demonstrate the efficiency of the electric car in suburban service as well as to show its proved status again as a town car. In a certain sense the run was not a formal test as it was conducted by the contestants themselves, but all the entrants agreed to the officials selected and there was a spirit of unity about the affair that augured well for general results.

All of the companies represented state that a single charge of electricity will prove sufficient to carry its cars from start to finish and leave a large amount in reserve. Several ladies figured as drivers and there were a number of feminine passengers in the contesting column.

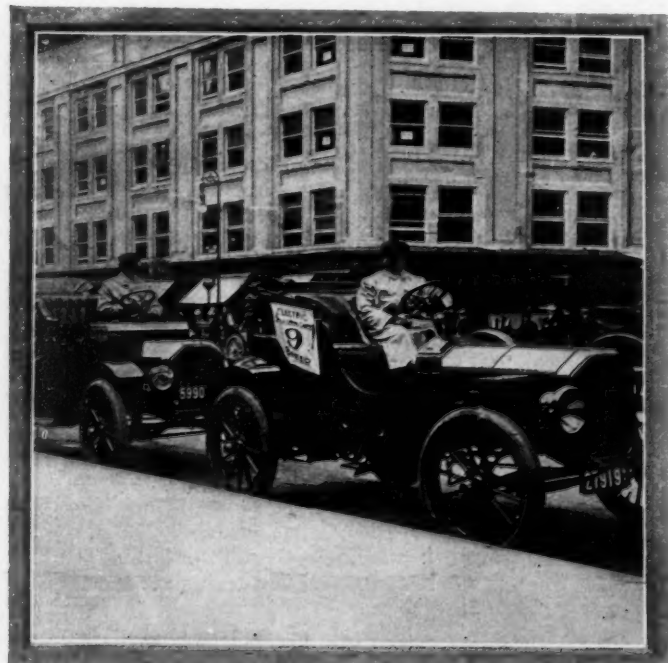


Fig. 1—Line-up of the Baker Electrics at the starting line

The entry list included the following:

- 1—S. J. Wise & Co., Rauch & Lang.
- 2—Mrs. C. Y. Kenworthy, Rauch & Lang.
- 3—C. Y. Kenworthy, Rauch & Lang.
- 4—Anderson Electric Car Company, Detroit Electric.
- 6—Anderson Electric Car Company, Detroit Electric.
- 7—Baker Vehicle Company, Baker.
- 8—Nathaniel Platt, Baker.
- 9—Miss Nita Sigura, Baker.
- 10—The New York Edison Company, Baker.
- 11—The New York Edison Company, Baker.
- 12—Paul Tietjins, Detroit Electric.
- 14—Walter Jones, Rauch & Lang.
- 15—Mrs. O. F. Alleman, Rauch & Lang.

## Saturday Matinee at Guttenberg

There will be a race meeting at Guttenberg race track Saturday afternoon in case the course is improved and made safe according to orders of the Contest Board. Seven events are carded, ranging from a dash of three miles for Class C cars of the smallest size to a ten-mile free-for-all, which may include the biggest road locomotives. Most of the events are open to amateurs.

The start of the first race has been set for 2:30 o'clock. The management has agreed to roll the course and oil it by this evening when it will be inspected.



Fig. 2—Albert Weatherbee, chairman of the tour, preparing to start.

## Syracuse Automobile Activities

SYRACUSE, N. Y., June 13—C. Arthur Benjamin, president of The Syracuse Automobile Dealers' Association, has received official word from the State Fair Commission that there will be an automobile race meet the last day of the 1911 exposition here.

June 13 is a day memorable at the plant of The H. H. Franklin Manufacturing Company, as it is the thirteenth anniversary of the delivery of the first Franklin air-cooled automobile, which was taken by S. G. Averell, of New York. Since this first sale some 10,000 of the air-cooled cars have been sold. The first car was of 10 horsepower and the motor was set crosswise in front of the dash. The drive was by chain. Averell used to drive the car in races and the air cooling arrangement attracted much attention. Experts declared that it never would be found practicable for high powered cars.

Thirteen has proved a lucky Franklin number. Arrangements for starting the business were completed on the 13th of the month, thirteen cars were turned out the first year and Franklin cars numbered "13" have won many national contests.

The Automobile Club of Syracuse has just completed its first road construction at a cost of \$50. Complaint was made to the organization that a small stretch of highway near the eastern

county line was in sad need of repairs. The club could not induce the county or town officials to do anything, so the club employed men to do the work and paid them therefor, a member donating the use of some teams. It was a marshy stretch of road in which a number of machines belonging to tourists had lately become mired.

### North Platte Highway Makes Progress

OMAHA, NEB., June 12—The interest which the people of Nebraska feel in the subject of good roads in general, and in a North Platte highway in particular, was clearly demonstrated last week, when the Omaha *World-Herald's* good road special automobile went over that highway, starting from Omaha, and going to North Platte.

At every town along the way, committees and lines of automobiles were on hand to greet the Inter-State "50" touring car, and wherever possible, short good roads meetings were held. In most cases, automobiles were waiting at the county line to escort

States to given an outing to orphans. Motor organizations throughout the country are now regarding this annual event as a fixture.

All of the children yesterday carried American flags and following the outing announced that they had had the time of their young lives.

### Orphans Ride in Philadelphia

PHILADELPHIA, June 12—As guests of the members of the Quaker City Motor Club and public-spirited citizens, about 1,000 kiddies from various institutions for poor children in Philadelphia started last Monday morning on a joy ride in the truest sense of that much-abused term, the objective point being Willow Grove Park. Nearly 100 cars were assembled at the Hotel Walton, and then rounded up the children, each car starting on the trip as soon as loaded.

### Engine Trades to Hold Convention

DETROIT, June 12—The National Gas and Gasoline Engine Trades Association will meet at the Hotel Pontchartrain, in this city, June 20-23, inclusive. A varied and interesting program will be presented during the three days and quite a feature will be made of the social side. The ladies who attend will be given special consideration. The business sessions will be comparatively short, and will be made up of brief, pithy addresses.

### Lozier Increases Capitalization

DETROIT, June 12—The Lozier Motor Company has authorized an increase of its capitalization from \$2,000,000 to \$3,000,000. Half of the additional issue will be of 7 per cent. cumulative preferred stock and the rest common. It is announced that the latter will not be issued at present. The purpose of the increase is to finance some contemplated enlargements of manufacturing plant.

### Nomenclature Report Delayed

On account of delay in coming to hand the report of the S. A. E. Standards Committee on Nomenclature will not appear in this week's issue of THE AUTOMOBILE.



Fig. 3—Number 5 contesting car, a Detroit Electric

the car across the county, some of these autos traveling a total of 100 miles while escorting them.

The good roads special was in charge of Dan V. Stephens of Fremont. The object of the trip was to stir up interest in this highway, and in good roads. The *World-Herald* has offered a cash prize of \$500 for the county having the best stretch of road on this highway, and \$300 for the township having the best road.

### Louisville Club Entertains Orphans

LOUISVILLE, June 12—Five hundred children from the different orphanages of the city were given a "joy ride" Saturday through Louisville's streets and parks and during the afternoon were the guests of the Louisville Automobile Club at Fontaine Ferry Park. All of the amusement devices were open to the children without cost. The orphans were treated to peanuts, candy, ice cream and other refreshments. Sixty-five cars assembled at Third avenue and Broadway, where the machines were assigned to the various orphanages. After each car had been loaded to its fullest capacity with little tots, they were driven to the park, the procession being led by B. B. Watts, chairman of the Orphan Day Committee of the Louisville Club. Mr. Watts was assisted by scores of members of the organization, which claims to have been the first auto club in the United



Fig. 4—Some of the Detroit cars in their places in line

## Savannah Makes Plans for Races

SAVANNAH, GA., June 12—Plans for the automobile carnival that will be held in this city next Fall have been tentatively formed, and the prospect is for such a season of automobile racing as has never been held before in the history of the industry. With the Grand Prize of the Automobile Club of America, the Vanderbilt Cup and the Savannah Challenge Cup, the Georgia city that has done so much for the automobile will have a trio of racing events the like of which have never been attempted anywhere.

The plans, as they have been framed, contemplate the running of the Vanderbilt Cup and Savannah Challenge Cup races Tuesday, November 28. There will be no racing on November 29, and the culmination will take place with the running of the Grand Prix on November 30, Thanksgiving Day.

In addition to these racing events, the club has made a proposal, more or less formally, to the Contest Board to run the Glidden Tour from New York to Savannah, reaching that city on November 24. While such an undertaking would appear to be somewhat late in the Fall for the tour, the club and the Savannah authorities are hopeful and enthusiastic over the prospect.

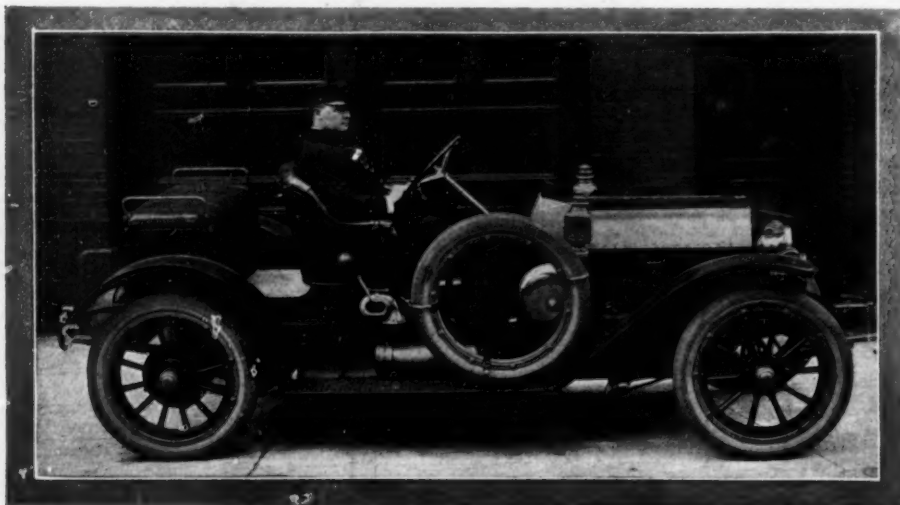
Accommodations for 50,000 visitors are being prepared and the committee has found that, aside from the new fifteen-story hotel now in course of construction, which will be ready for business in the Fall, that the new hotel at Tybee, 18 miles from this city, will be available, in addition to the regular hotel equipment and the private homes that were thrown open for visitors last Fall.

## Basle Meets His End at Hawthorne

Marcel Basle, one of the best known professional automobile drivers in the United States, was killed at Hawthorne race track Saturday while driving a racing car. Basle was a veteran, having achieved quite a reputation in France prior to coming to this country. He was brought over to act as mechanic for Frank Croker in a Simplex car entered in the first Vanderbilt Cup race and since then has been prominent in long distance events of various kinds.

## Maryland Club to Climb July 1

BALTIMORE, MD., June 12—The Automobile Club of Maryland will hold a hill climb on the Belvidere Hill, Mount Washington, on July 1. It was originally intended to hold this event in the Fall but the date was changed at the urgent request of members of the club and others interested in the affair.



Chief O'Donnell, of the St. Louis Salvage Corps, in his 1911 Moon Model "30" Roadster

The climb will be under sanction. It is further announced by the hill climb committee that if enough entries are secured among the members and others interested in the climb, an individual owners' class will be established. The committee comprises Joel G. Nassauer, chairman; Edgar F. Dodson, Asa B. Gardiner, Jr., and H. M. Luzius.

## General Motors Pays First Note

DETROIT, June 12—Announcement has been made by the General Motors Company of the payment of \$500,000 of its large loan, negotiated late last Fall. On October 1, \$1,000,000 more will be due. This is already practically provided for out of the profits of the Cadillac, Oldsmobile, and other properties.

In general it is noticeable that the tone of business is much stronger than was the case a year ago. While some of the Detroit factories have not produced as many cars as they did last year, none is confronted with the conditions which prevailed in the face of the tremendous outputs of 1910.

There is considerable expansion to be noted at the Detroit plants. The Packard addition is well under way. The Studebaker Corporation is building a big, three-story addition to its Flanders "20" plant. Building operations are in progress at the Cadillac factory, where the former Woodward avenue car barn, recently added to the company's holdings, is being transformed into a serviceable factory adjunct. At nearly all the factories additional room for manufacturing operations is being provided.

Demand continues excellent, particularly for cars of the light 20-30-horsepower type. The Ford Motor Co. recently made the interesting statement that it would not be difficult to busy the entire factory with the orders for cars from their agents outside the United States. The Ford was one of the first, if not the very first to consistently encourage the export trade and manufacturers who have gone into this branch more recently, regard the success of the pioneer as most encouraging.

Motorists of Detroit continue their assaults on the recent radical traffic law passed by the council. Two meetings have been held at which these protests were placed before the council's committee in a formal way.

## T. C. A. Establishes Four Branches

Following its recently conceived plan of establishing branch headquarters at various centrally located points, the Touring Club of America has installed four branches in the most picturesque spots in New England.

The new branches with their locations are: The Naugatuck Valley branch in the Hotel Elton at Waterbury; Connecticut Valley Branch in the Hotel Kimball at Springfield, the White Mountain branch in the Mount Washington Hotel, Bretton Woods, N. H., and the Berkshire Hills branch at the Hotel Wendell, Pittsfield, Mass.,

The following advisory board for the Washington, D. C. branch of the club has been named: W. Piatt Andrew, Major Archibald W. Butt, General Clarence R. Edwards, Dr. Logan Walker Page, E. B. McLean, Rudolph Kauffman, Samuel A. Luttrell, Cuno H. Randolph, J. M. Stoddard, P. S. Johnson, M. T. Pollock, C. J. Bell, Benjamin Woodruff, Dr. Ralph Jenkins and George W. White. This list contains the names of representative citizens of Washington and officials of various civil and military departments.

## Quakers See 300 Trucks on Parade

PHILADELPHIA, June 12—A forcible and convincing demonstration of the adaptability and practical uses of the motor-driven commercial vehicle in the transportation field was given to-day when over 300 trucks, ranging from mammoth conveyances carrying heavy loads to the smallest and lightest wagons, with every conceivable type represented between the two extremes, progressed through the principal business thoroughfares of Philadelphia, culminating in a monster exhibition of the participating vehicles at the Belmont Driving Park, Narberth.

The occasion was an industrial parade conducted under the auspices of the *Philadelphia Inquirer*, with the co-operation of the Philadelphia Auto Trade Association.

Originally conceived with the sole purpose of being an object lesson in the all-around desirability of power-driven vehicles for the delivery of merchandise, the demonstration more than fulfilled expectations. It was by far the largest number of commercial vehicles ever assembled at one time, in Philadelphia at least, and representative of over 50 different makes.

In order not to add further to the congestion in busy Market street, the vehicles were classified according to size, capacity, etc., and formed on Spring Garden, Fifteenth, Sixteenth and Seventeenth streets early and proceeding to the official starting point in front of the Inquirer Building, Eleventh and Market streets, at designated intervals. Originally scheduled to start at 10 o'clock, it was nearer 11 when the head of the parade approached the starting line. However, considering the vast number of entries, there was very little confusion, and once under way, no further hitch was experienced.

The parade was headed by a platoon of mounted police and the pathfinding Kisselkar. The cars were classified into four divisions, each division being headed by a band seated in huge trucks. Division No. 1 contained trucks with a capacity of three tons and over; Division No. 2, over 1500 pounds and less than 3 tons; Division No. 3, 1500 pounds and less; Division No. 4, electric vehicles. As a general rule, the cars were those taken out of active service, to be returned to duty again upon the conclusion of the run. There was a liberal sprinkling of manufacturers' models.

Merchants located along the line of march, many of whom had their trucks in the long line, decorated their places of business with flags and bunting, the principal thoroughfares taking on the appearance of a holiday, and thousands of spectators lined the sidewalks.

## Velie's Fine Chicago Home Burned

CHICAGO, June 12—Fire broke out in the local salesrooms of the Velie Motor Car Company, 1615 South Michigan avenue, at 4 o'clock Saturday afternoon, caused, it is said, by some one stepping on a match, which ignited waste paper lying on the third floor. The third floor of the building was burned before the flames were subdued, causing a damage estimated at \$60,000. Seven cars were destroyed, while 20 others on the second floor were damaged by water.

## Downs Goes to Autocar Co.

Merle L. Downs, who for a number of years has been Secretary of the Show Committee of the A. L. A. M. Shows in Madison Square Garden, has associated himself with the Autocar Company, of Ardmore, Pa. Mr. Downs will devote a large part of his time to the study and development of the production and transportation end of the business.

## Worm-Drive Truck Makes Good Run

Loaded with five tons of lubricating oils, in barrels, a Pierce-Arrow five-ton truck made the run from New York to Boston in 20 hours flat, according to those who made the trip. The elapsed time was from 2 o'clock Friday morning to 3 o'clock Saturday afternoon, but between these hours time was taken out for demonstrations at New Haven, Hartford and Springfield, and the night was spent at the latter place.

The truck used for the trip was one that had been in use for a month for demonstrating purposes in Chicago, Philadelphia, New York and Newark, and on the Monday following the run it was due in Boston. The Standard Oil Company was interested in the matter of inter-city hauling and a load, consigned to a Boston firm, was secured from it.

The schedule of the trip was:

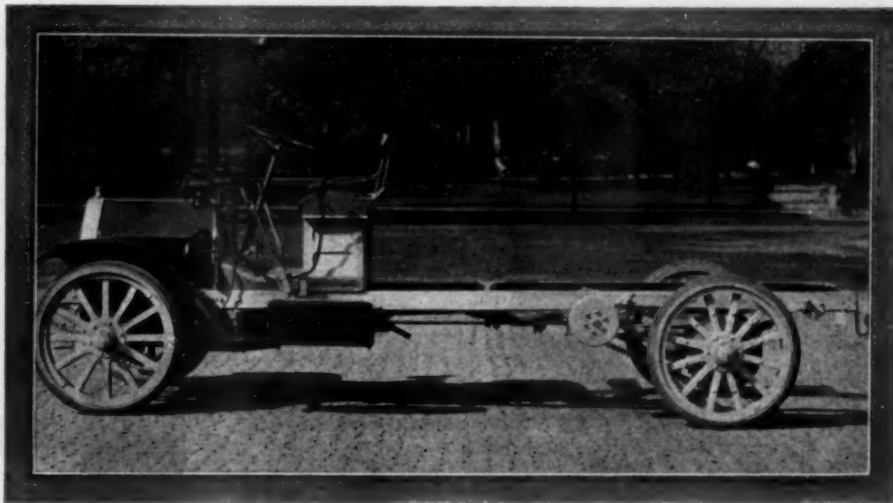
	Running Time.
Left New York.....	2:05 A. M.
Arrived New Haven.....	8:27 A. M. 6:22
Left New Haven.....	9:30 A. M.
Arrived Hartford.....	12:25 P. M. 2:55
Left Hartford.....	5:00 P. M.
Arrived Springfield.....	7:15 P. M. 2:15
Left Springfield.....	5:27 A. M.
Arrived Worcester.....	9:50 A. M. 4:13
Left Worcester.....	10:45 A. M.
Arrived Boston.....	3:00 P. M. 4:15

Total running time.....20:00

This would make the average speed for the entire run 12.1 miles per hour, the speedometer showing that the distance covered on the road was 242 miles. During the trip 51.5 gallons of gasoline were used, which would give an average mileage of 4.7 miles to the gallon. Fully 20 miles were covered in demonstrating trips in the cities visited and the motor was allowed to run at least an hour in the different cities where stops were made. In all, six pints of lubricating oil were used, an average of 40 miles to the pint.

## Cole Company to Enlarge Plant

INDIANAPOLIS, June 12—Plans are being perfected for the construction of an enlarged manufacturing plant for the Cole Motor Car Company. The factory will not be removed from this city. Financial arrangements covering the building plan have been made. Definite announcement of the date upon which building will commence has not been made. It is planned to increase the output materially next year and to specialize in the manufacture of pleasure cars as in the past. The company has turned out a few trucks.



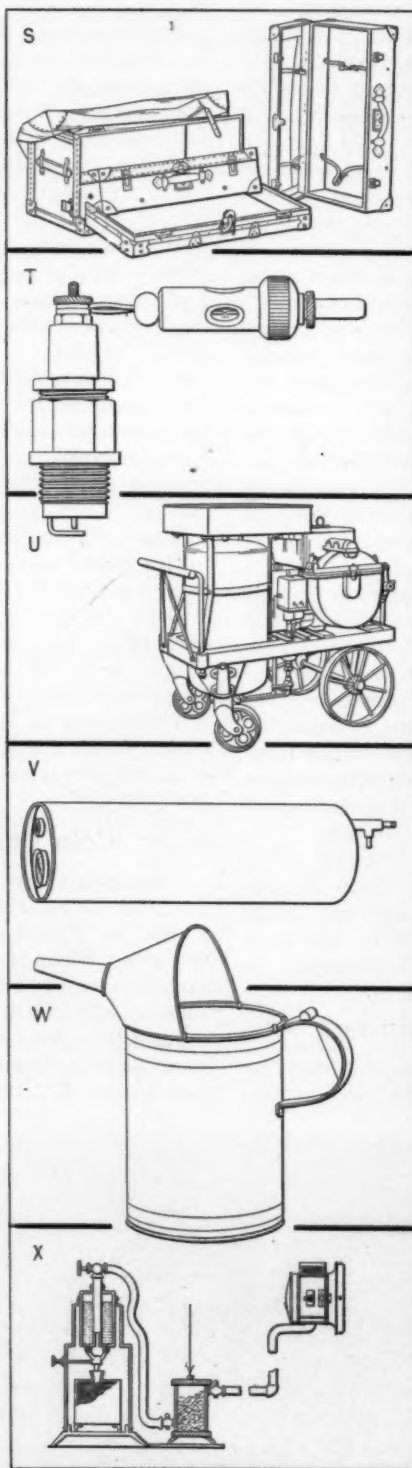
One-and-a-half-ton furniture van chassis turned out by the Federal Motor Truck Company, Detroit, Mich.

## Seen in the Show Window

**T**OURING in the modern sense means a thousand little or big articles which cannot be left behind, and therefore have to be taken in the automobile. The needs of the autoist have been well grasped by the makers of the Kamlee auto trunk (S), the Kamlee Company, 341 Milwaukee street, Milwaukee, Wis. The trunk fits under the tonneau of any make of car, and is equipped with three standard sized suit cases, which are obtained by lowering the patent drop front, it being unnecessary to take the trunk out of its place. The material used for the production is three-ply bass wood veneer, and the lining genuine Irish linen, which is covered with heavy black enameled duck. A guarantee goes with every trunk.

**M**ISFIRING may be due to more than one cause, and its reason having once been found out remedying the trouble will be a comparatively easy undertaking in the majority of cases. A most troublesome phase in the whole process is not infrequently the detection of the one spark plug which misses fire, after the symptoms of misfire have been noticed by the automobilist, and spark plug manufacturers are ever striving to improve the situation by devising means for easily detecting which plug is at fault. The Misskip detector, product of the Champion Spark Plug Company, 601 Jefferson avenue, Toledo, which is herewith illustrated, is applied to the terminal screw of a spark plug in the manner shown at (T), and if a spark plug is not functioning properly the current short-circuited into the detector will produce no spark between the two points seen in the opening of the insulator handle.

**G**ARAGES have long ago taken to air compressors, and progress in this field is along lines of increasing simplicity, one of the more recent designs being of the portable type which is shown at (U). The National outfit comprises an electric motor for direct or alternating current, furnished in accordance with the buyer's wishes, an air-compressing pump and an air storage tank, so that an abundance of compressed air may always be at hand. The outfit is compact and of solid construction, and is ever ready for operation, as it may take current by means of a standard lamp socket. It may be used for cleaning cushions, upholstery, etc., as well as for the common work of inflating tubes, so that it is an instrument of all-around usefulness. It is made by the National Brake & Electric Company, Milwaukee, Wis.



(S) The Kamlee auto trunk, handy for the tourist  
(T) Misskip detector, to locate spark plug that is missing fire  
(U) The National portable air compressor, for garage use  
(V) Simplicity and beauty combined in the Meteor gas tank  
(W) The Dover combination radiator filler and filter  
(X) Showing the operation of the Wald Acetylene gas apparatus

**T**HE practice of using acetylene gas for the headlights on automobiles is continually increasing, and the growing demand necessitates an enlarged output of machinery for producing and storing illuminating gas. Various makes of gas tanks are now on the market, being the products of corporations who use modern methods in their manufactories and supply their tanks to their representatives all over the country, where automobilists are supplied with the acetylene. Progress in this field is chiefly in the nature of refinement of details, and since appearance is not unimportant, and tastes vary, the Meteor tank shown at (V) was given a nickel cover. The tank, which is of the regular size, is made by the Meteor Auto Tank Co., Middlehaddam, Conn.

**T**HE importance of properly cooling an internal combustion motor is generally recognized, and a well-functioning radiator being installed on a car all that remains is to get the chauffeur to keep it in shape and utilize it to its fullest extent by always keeping it full of clean, soft water. Renewing this medium is uncomfortable at times, and the use of a combined vessel and funnel as illustrated at (W) will enhance the probability of the driver's periodical looking after the radiator. The filler holds six quarts, and is provided with a strainer. The material is galvanized, rust-proof steel, and while the capacity of the vessel is not a very large one it is reasonable to expect that, considering the ease of handling, the chauffeur will always refill the radiator when several quarts of water have been evaporated. The Dover Stamping & Manufacturing Company, of Cambridge, Mass., are the producers of this appliance.

**C**LEAN acetylene is an indispensable condition for good lighting, and there is only one way of freeing a gas from foreign, especially solid, matter, which is by passing the gas through a suitably contrived filter. This sort of apparatus is often thought of as being voluminous and heavy, but the Wald Manufacturing Company, of Sheboygan, Wis., has listed among its products a compact and nevertheless efficient filter for illuminating gas on automobiles. As may be seen by looking at the illustration (X) the operation of the filter is as follows: The gas which is generated in the lower compartment of the higher tower rises through the vertical pipe and then descends into the small cylinder, passing through a layer of solid material where all inherent moisture and small solid particles are withheld.